# THE MARINE POWER PLANT

# THE MARINE POWER PLANT

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# PREFACE

The purpose of this book is to bring before the student the thermodynamics of the marine power plant, the types of machinery used for ship propulsion, and to give him a comprehensive idea of the layout and function of the various pieces of auxiliary machinery.

The book makes no pretenses at being an exhaustive treatise. It is intended as a first book in marine engineering. At Lehigh University the study of the marine power plant as presented in this book, is preceded by a course in thermodynamics and followed by a summer at sea and by a more thorough and detailed study of marine engines, turbines and Diesel engines.

The thermodynamic and economic features of the power plant have been accentuated throughout the book. Very little attention has been given to mechanical details and all pure descriptive matter has been reduced to a minimum. Details can be better learned under actual operating conditions on shipboard than from the inadequate treatment in a text book.

A short chapter on thermodynamics has been added as a review for the engineering student and also as a foundation study for others who may study the book. Complete calculations for the sizes of the boilers and auxiliaries of a typical plant are given in Chap. XIX. It is believed that this is the first time such calculations have appeared in print.

A special feature of the book is the comparison of the various types of machinery used today for ship propulsion which is concluded with a table showing an unbiased comparison of seven types of propelling machinery.

While the book is intended primarily for the students of naval architecture, marine engineering, and ship operation, it is believed that it will bring before the sea going engineer and ship owner a better understanding of the many types of propelling machinery and auxiliaries used today.

L. B. Chapman.

Bethlehem, Penn.,
*June, 1922.*

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L. B. CHAPMAN.

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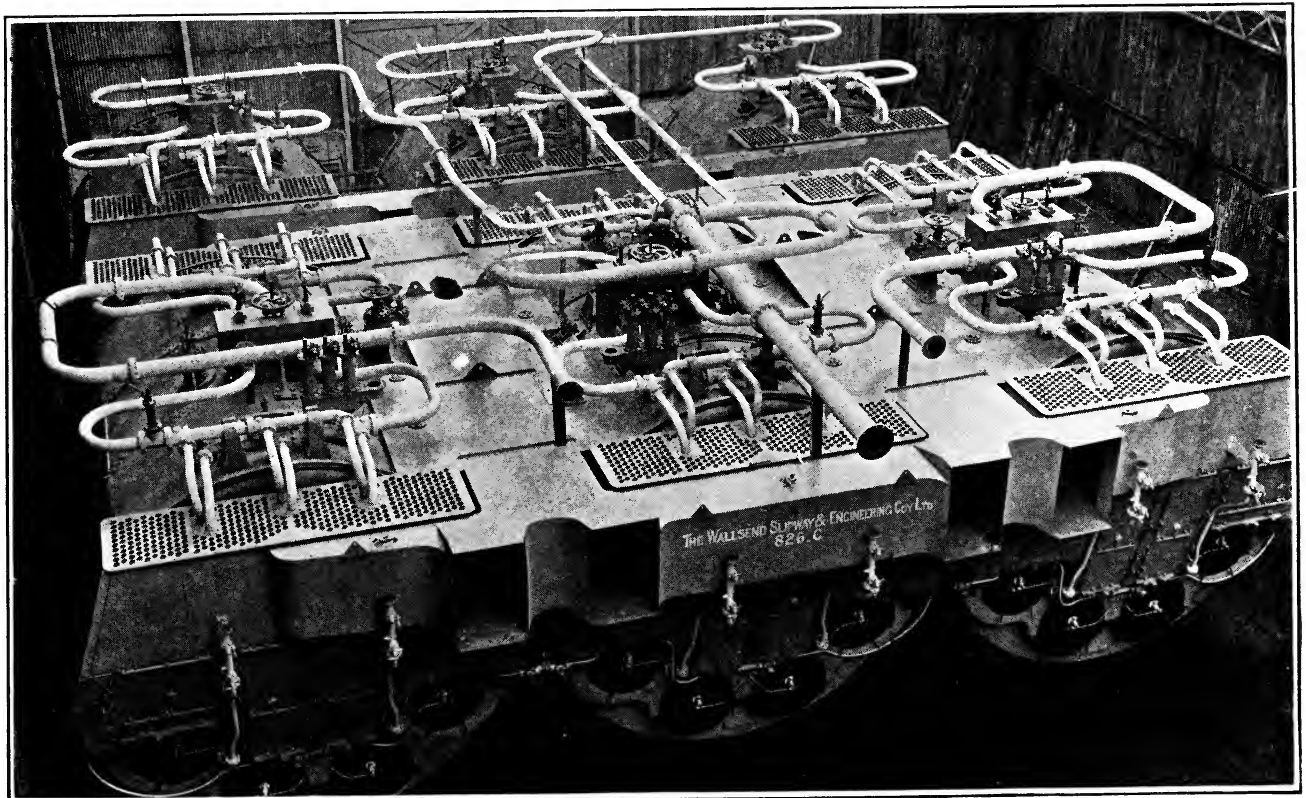


(Frontispiece)

Boiler installation and steam-pipe arrangement S.S. "Laconia".

# THE MARINE POWER PLANT

## CHAPTER I

### INTRODUCTION

**1. Types of Machinery.**—In 1905 practically the only type of machinery installation used on board merchant ships was the reciprocating engine, and Scotch boiler in which coal was the fuel universally used. In 1922, 16 years later, marine engineering is undergoing radical changes and is still in a state of flux. Many new types of propelling machinery are in use and still newer types are being strongly advocated. While the reciprocating engine and Scotch boiler are still used to a large extent, the newer types are fast superseding this older type. Coal fuel has practically been replaced by fuel oil.

Today we find the geared turbine and water-tube boiler used almost as much as the reciprocating engine and Scotch boiler. In addition, combination machinery, steam turbines, electric drive, Diesel engines, and Diesel-electric drive are being widely installed on many types of ships; and a combination type of steam and Diesel engine, known as the Still engine, is passing the experimental stage.

With all these various types available for ship propulsion the marine engineer and shipowner has a more interesting and difficult task in deciding on the proper type of machinery than was the case 16 years ago. All of these types will be fully treated in the following pages and the advantages and disadvantages of each type fully discussed.

**2. Comparison of Land and Marine Installations.**—The problems confronting the designer of a marine power plant are far more difficult than those confronting the designer of a power plant on shore. The marine engineer and stationary engineer have many problems bearing on efficiency, economy and costs that are almost identical, and in many ways the power plant

on shipboard and on shore are much alike. The marine engineer, however, has limitations of weight and space that are not met by the engineer on shore.

For high speed ships it is vital, in order to reduce the resistance, that the displacement be kept as small as possible. This necessitates propelling machinery of light weight per horsepower. The natural way to reduce the weight of machinery for a given horsepower would be to increase the revolutions. High revolutions of the propeller, however, are not compatible with good propeller efficiency and high propeller efficiency is necessary in order to keep the power of the ship at a given speed as low as possible. Thus the designer is confronted at the outset with two restricting conditions that conflict with each other.

While low steam and fuel consumption are just as important on shore as on shipboard from the viewpoint of economy, the marine engineer has additional incentives for low fuel consumption besides cost of fuel. A reduction in the fuel consumption of a ship's power plant reduces the weight and space occupied by fuel in the bunkers and consequently allows increased space and weight for cargo which in turn results in a greater earning capacity of the ship. If the steam consumption of the propelling machinery can be reduced, the size and weight of boilers, piping and main auxiliaries can be reduced with a further increase in the earning capacity. The problem of power plant design for a ship of high speed is far more difficult than that for the slow speed cargo ship; yet there are many features such as low propeller r.p.m., and space requirements that need careful attention with the latter type.

The necessity for low steam consumption, light machinery and low propeller revolutions have been the cause of the introduction of fuel oil, light water-tube boilers, geared turbines and electric drive. The low fuel consumption of the Diesel engine has caused it to be adopted for low powered cargo ships, notwithstanding its greater initial cost and increased weight over steam machinery.

The average fuel consumption and weights of the various types of machinery used today are given in the table on the following page.

**3. The Elementary Steam Power Plant.**—Practically all marine power plants are condensing as it is necessary to save the

condensed steam for feed water and surface condensers are used entirely for ships navigating in salt water.

A diagrammatic layout of a simple plant is shown in Fig. 1. The steam is generated in the boiler by absorbing heat from the coal; the steam leaves the boiler with the heat contents $x_1r_1+q_1$ per pound and enters the engine or turbine through the throttle valve. In the engine, mechanical work is done at the expense of the heat in the steam. The heat left in the steam after expansion in the prime mover is rejected to the condenser. Here the latent heat in the steam, $x_2r_2$ is absorbed by the water circulating through the condenser and discharged overboard. The air pump removes the condensed steam from the condenser and discharges it with the heat contents of $q_2$ to the feed tank. The boiler feed pump takes the feed water that collects in the feed tank and delivers it under pressure to the boiler.

Table I

| | Boiler Press. | Vacuum | Propeller r.p.m. | Total Mach. lbs./hp. | Fuel lbs./hp. per hour |
|---|---|---|---|---|---|
| Reciprocating engines, Scotch boilers and coal. | 200–250 | 25″–27″ | 75–85 | 500 | 1.6 |
| Reciprocating engines, Scotch hoilers and oil. | 200–250 | 25″–27″ | 75–85 | 500 | 1.2 |
| Geared turbines, water tube boilers and oil. | 250–300 | 28½″ | 90 | 300 | 1.00 |
| Geared turbines, express type boiler and oil. | 250 | 28½″ | 450 | 42 | 0.95 |
| Four-cycle Diesel engines | ...... | ........ | 85 | 650 | .42 |
| Four-cycle Diesel engines | ...... | ........ | 115 | 600 | .42 |
| Two-cycle Diesel engines | ...... | ........ | 85 | 550 | .44 |
| Still engine | ...... | ........ | 125 | ...... | .375 |
| Turbo-electric drive and w.t. boilers. | 250–300 | 28½″ | 90–100 | 350 | 1.00 |
| Diesel electric drive | (250 r.p. | m. Gen.) | 90–100 | 450 | .53 |

All weights except that of the reciprocating engine installation are based on shaft horsepower. Superheat is not used in any of the above installations.

The object of the condenser is two-fold. It saves the condensed steam so that it can be used again in the boiler; and it

reduces the back pressure on the engine by means of the vacuum produced by the condensed steam. This reduced back pressure allows a greater expansion of the steam in the prime mover and a larger amount of the heat in the steam is converted into mechanical work. A complete diagram of a ship's power plant showing all the auxiliaries, feed water heaters, etc., is shown in Fig. 105.

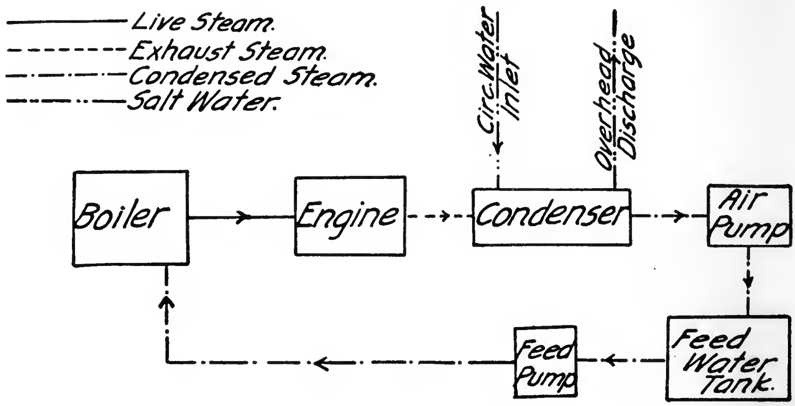


FIG. 1.—Diagram of elementary power plant.

**4. Overall Plant Efficiency.**—If we use the values for average fuel consumption given in the above table and assume the calorific value of coal as 14,300 b.t.u. per pound and fuel oil as 18,800 b.t.u., we have for the overall efficiencies of the power plants the following values:

$$\text{Reciprocating engine and coal} = \frac{2545}{1.6 \times 14{,}300} = 11 \text{ per cent.}$$

$$\text{Geared turbine and oil} = \frac{2545}{1.00 \times 18{,}800} = 13.5 \text{ per cent.}$$

$$\text{Four-cycle Diesel engine} = \frac{2545}{.42 \times 18{,}800} = 32.2 \text{ per cent.}$$

The above figures show how small a percentage of the heat in the fuel is utilized for useful mechanical work. For the steam installations between 89 per cent and 86 per cent of the available heat is lost while with the Diesel this loss is greatly reduced, being only about 68 per cent. The following chapters will be

devoted to a large extent to the study of these losses and the means adopted to increase this overall plant efficiency and thereby reduce the fuel consumption.

The losses for the geared turbine and water-tube boiler are roughly as follows:

| | | Per Cent |
|---|---|---|
| Boiler losses: (boiler efficiency 75 per cent) | | |
| Heat lost up stack | | 12.0 |
| Incomplete combustion, moisture in fuel, etc. | | 7.0 |
| Radiation | | 6.0 |
| Engine losses (friction, leakage, radiation) | | 2.0 |
| Rejected to condenser cooling water | | 49.0 |
| Pipe radiation and leakage (total plant) | | 1.0 |
| Used by auxiliaries | | 18.0 |
| Returned by auxiliary exhaust to feed water | 9.0 | |
| Total net loss | | 85.0 |

In a Diesel engine installation, in which all the auxiliaries are driven by the main engine, the losses are approximately as follows:

| | Per Cent |
|---|---|
| Loss in jacket circulating water | 10 |
| Rejected in exhaust | 30 |
| Radiation and minor losses | 1 |
| Engine friction and auxiliary power | 27 |
| Total loss | 68 |

# CHAPTER II

## REVIEW OF THERMODYNAMICS OF STEAM

It is assumed that the student has made a study of the thermodynamics of the steam and gas engine and is familiar with the use of the steam tables and diagrams. The following chapter has been prepared as a brief review of the more important features of thermodynamics that are necessary for a study of the marine power plant. The thermodynamics of the steam turbine is given full treatment in Chap. X.

**5. The Generation of Steam.**—For engineering purposes steam is generated by the combustion of coal or fuel oil. The feed water is injected into the boiler at some temperature above 32°F.[1] The heat contained in this water above 32° is expressed in b.t.u.'s per lb. by the symbol $q_2$. The water in the boiler now absorbs heat produced by the combustion of the fuel and its temperature rises until it reaches the boiling point at some predetermined pressure. The heat in this water in the boiler expressed in b.t.u.'s per lb. above 32°F. is $q_1$ (the heat of the liquid). The heat added to the water thus becomes $(q_1 - q_2)$ b.t.u. per pound.

In order to understand clearly the phenomenon that is taking place in the boiler and the application of the steam tables, it is convenient to assume that the water in the boiler is under an initial air pressure of $P_1$ lbs. per sq. in. As steam begins to form in the boiler first the air and then the steam is drawn off at the same rate as the steam is generated. Thus the pressure in the boiler is always constant.

As further heat is added to the water which has already reached the boiling temperature, the water begins to boil at a constant temperature corresponding to the pressure $P_1$ and steam is formed. The heat absorbed in changing a pound of water at the boiling point into steam is known as the heat of vaporization and is expressed by the symbol $r_1$. If the boiling takes

[1]The steam tables are based for convenience on the heat contents above 32°F.—in other words water at 32° is assumed to have zero heat contents.

place in a violent manner or in too confined a space a small part of the water, perhaps 1 or 2 per cent is thrown off and does not form steam but is held in suspension in the steam. Obviously, if 2 per cent of every pound of water is thus thrown off with the escaping steam bubbles this 2 per cent does not absorb the heat of vaporization $r_1$ but contains only the heat $q_1$. In such a case the heat absorbed by the water per pound is $.98r_1$. The heat of vaporization is expressed as $x_1r_1$ where $x_1$ represents the percentage of water actually turned into steam. This factor $x_1$ is called the quality of the steam. When the steam contains moisture it is known as wet steam; but if all the water in every pound is turned into vapor and the quality is 100 per cent the steam is said to be *dry* or *saturated* steam. The unit in all steam computations is the pound and the values given in the steam tables are b.t.u.'s per lb. above 32°F. Each pound of water converted into wet steam contains $(q_1+x_1r_1)$ b.t.u.'s. The total heat of dry steam is expressed by $\lambda$ and is equal to $(q+r)$. If the steam is not saturated but with quality $x_1$ the heat contents are less than the total heat of dry steam. The heat contents are then expressed by $H_1=q_1+x_1r_1$. The heat absorbed in the boiler becomes $x_1r_1+q_1-q_2=H_1-q_2$ where $q_2$ is the heat contained in the feed water. In this book $H$ will always represent the total heat of steam but in many cases the steam will be wet. It should be clearly distinguished from $\lambda$ which always is the total heat of saturated or superheated steam taken directly from the tables.

If the steam is now withdrawn from the boiler and subjected to further heating more heat is absorbed and the steam becomes superheated. The total heat contents per pound now becomes $H_1=r_1+q_1+Ct_s$, where $t_s$ equals the number of degrees of superheat added above the boiling temperature and $C$ is the specific heat of superheated steam.

The steam tables also give the specific volume, or the volume occupied by dry steam under various pressures and degrees of superheat. The symbol used for this is $s$. If the steam is wet with a quality of $x$, the specific volume becomes $xs$. For use in solving problems involving intrinsic energy of steam such as adiabatic expansions the heat of vaporization $(r)$ is split up into the heat equivalent of external work $Apu$ $(Apxu)$ and the heat equivalent of internal work, $\rho(x\rho)$. Hence $r=\rho+Apu$.

*Intrinsic Energy.*—The heat energy stored within the steam is termed the intrinsic energy. This does not represent the total heat of the steam

$(xr+q)$ because during the process of vaporization and the accompanying increase in volume some of the heat energy was used in doing external work in pushing back the surrounding media and this portion of the energy $Apu$ is considered as stored in the surrounding media. The intrinsic energy of steam thus becomes $x_1r_1 - Ap_1x_1u = x_1\rho_1 + q_1$ or expressed in ft. lbs.,

$E = 1/A(x_1\rho_1 + q_1)$ where $A$ is the reciprocal of the mechanical equivalent of heat, $p$ is the pressure of the steam and $u$ is the increase in volume of one pound of water in passing from water into steam.[1]

**6. Entropy.**—Entropy is a valuable property of steam and is useful in solving many problems in steam engineering. All our work with this property is limited to changes in entropy rather than with the absolute amount. If $d\phi$ represents an infinitesimal change in entropy, we can define this change in entropy as $d\phi = dQ/T$ where $dQ$ represents an infinitesimal change in the heat contents and $T$ the constant absolute temperature at which this change takes place. Therefore, the common definition of entropy becomes,

$$\phi_1 - \phi_2 = \int dQ/T.$$

As just pointed out all our study with entropy applies to *changes in entropy;* yet we are accustomed to speak of *total entropy* meaning thereby the entropy change above 32°F.

An adiabatic change of gas or vapor is one under such conditions of insulation that no energy, in the form of heat is given up nor none received during the change. Thus for a reversible adiabatic, $dQ = 0$ and from our definition of change in entropy given above,

$$\phi_1 - \phi_2 = 0$$

We can thus make the important statement that during a reversible adiabatic change (expansion or compression) there is no change in entropy.

A convenient way of thinking of entropy in the early stages and securing some tangible grasp of its significance, is to consider entropy that property of steam which is constant during a reversible adiabatic expansion, in a similar way in which temperature (a familiar property) is constant during an isothermal expansion. This of course is not a true and complete definition of entropy but one often found helpful in the early study of thermodynamics.

[1]For a full explanation of the above the student is referred to Hirshfeld and Barnard's "Elements of Heat Power Engineering," Chap. II.

A true understanding and familiarity with entropy can only be acquired by the solution of problems involving its application.

The steam tables give values of $\theta$ entropy of the liquid, and $r/T$, entropy of vaporization. The total entropy of the steam is, $\phi=\theta+\frac{xr}{T}$. The total entropy of superheated steam can also be found from the steam tables.

**7. The Temperature—Entropy (T$\phi$) Diagram.**—One of the valuable uses of entropy is its application in the $T\phi$ diagram. Here absolute temperatures are laid off as ordinates and values of entropy as abscissas. As entropy is zero at 32°F. or 492° absolute, our starting point for liquid at 32°F. is represented at O on the chart, Fig. 2. The entropy of the liquid for various

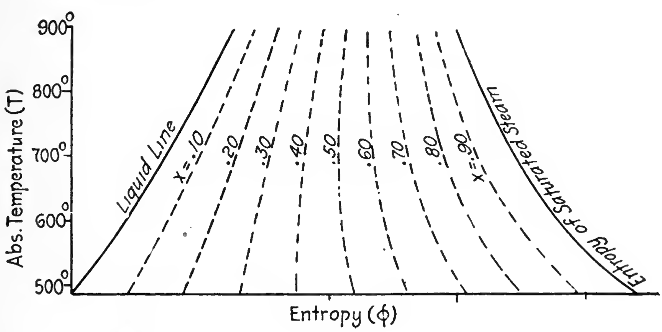


FIG. 2.

temperatures is now plotted on the chart, giving the liquid line In a similar manner the entropy of saturated steam ($r/T$) is plotted for various temperatures from the data given in the steam tables. This gives the line of "entropy of saturated steam." Intermediate between these lines are various intermediate conditions of $xr/T$ at the various temperatures. Quality lines are constructed giving the quality from 0 to 100 per cent.

From our equation for entropy $d\phi=dQ/T$, $dQ=Td\phi$ or the change in heat, $Q_2-Q_1=\int_1^2 Td\phi$. The right-hand member in the above expression is at once recognized as the expression for area under a curve. If we take some point as (2) Fig. 3 at temperature $T_1$, the change in entropy between point (8) and point (2) is $d\phi$. The heat added is $Q_2-Q_1=\int_2^8 Td\phi=\int_2^8 Td\theta$, where

$\theta$ = entropy of the liquid. This expression is represented by the area *9-2-8-5* in Fig. 3. Therefore the area on the $T\phi$ diagram represents heat change as in a similar way the area on the $PV$ diagram represents work. This area *9-2-8-5* must extend down to absolute zero for it is represented by $\int T d\theta$ where $T$ is absolute temperature and $d\theta$ is the change entropy.

Obviously, area *0-1-8-5* representing $\int_0^5 T d\theta$ must represent the heat of the liquid $q$. Likewise area *8-3-4-5* = $(r/T \times T) = r$ represents the heat of vaporization. Area *0-1-8-3-4* represents the total heat $(q+r)$. If the steam is not entirely dry and saturated $r$ becomes $xr$ and $r/T$ becomes $xr/T$. The area *8-6-7-5* represents $xr$ under these conditions.

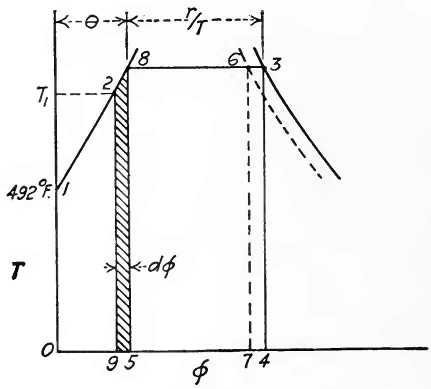


FIG. 3.

**8. Mollier Diagram.**—Another very useful diagram is the Mollier or $H\phi$ diagram. The data from the steam tables is plotted on this chart with total heat contents as ordinates and total entropy as abscissas. Lines of quality and superheat (which is quality above the saturated condition) and steam pressures are plotted on the Mollier diagram. Thus the total heat contents and total entropy for any condition of steam pressure, quality or superheat can be read off this chart at once, although not with the same degree of accuracy as in the tables.

**9. Adiabatic Expansion.**—As already mentioned (Article 6), an adiabatic expansion or compression of steam or gas is one in which there is no heat received or given off to the surrounding media; during a reversible adiabatic change, entropy is constant.

The most common reversible adiabatic expansion is the expansion of steam in the cylinder of an ideal reciprocating steam engine.

In Fig. 4, if dry steam at pressure $P_1$ corresponding to temperature $T_1$ is expanded adiabatically to pressure $P_2$ corresponding to temperature $T_2$, it will be represented on the $T\phi$ diagram by a vertical line extending downward from point *1* on the dry steam line. Entropy $\phi$ is constant and the temperature is reduced, therefore the adiabatic expansion is represented by the vertical line *1-2*. At condition *1* the steam was dry but as it expands at constant entropy the steam becomes wet. This is clearly shown in Fig. 4 where the line *1-2* cuts lines of lower and

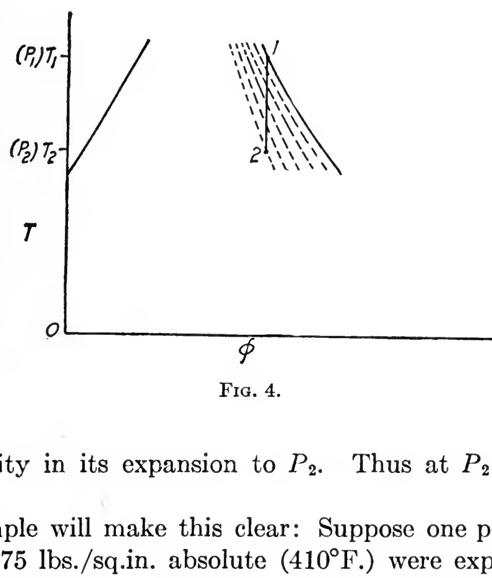


FIG. 4.

lower quality in its expansion to $P_2$. Thus at $P_2$ the quality is $x_2$.

An example will make this clear: Suppose one pound of dry steam at 275 lbs./sq.in. absolute (410°F.) were expanded adiabatically to 6 lbs./sq.in. absolute (170°F.). Noting the initial condition on Peabody's $T\phi$ diagram, we have $\phi_1=1.52$ $t_1=410°$ $x_1=1.00$. As the entropy does not change $\phi_1=\phi_2=1.52$ and running down along the line $\phi=1.52$ to $t_2=170°$ we have $x_2=.805$. The same results can be obtained from the Mollier diagram by looking up the intersection of the 275 lbs. line with the saturated steam line. The entropy is found to be 1.52 and running down the line $\phi=1.52$ until it intersects the 6 lb. pressure line, $x_2$ equals .805 as before.

The quality after the expansion can also be found by equating the entropy expressions for conditions $P_1$ and $P_2$.

$$\phi_1=\phi_2. \qquad \theta_1+r_1/T_1=\theta_2+x_2r_2/T_2.$$

Here, $\theta_1$, $r_1/T_1$, $\theta_2$ and $r_2/T_2$ can be looked up in the steam tables and $x_2$ the only unknown obtained.

$$\theta_1=.5771 \qquad \theta_2=.2476$$

$$r_1/T_1=.9429 \qquad r_2/T_2=1.5812$$

$$\theta_1+r_1/T_1=\theta_2+\frac{x_2r_2}{T_2}=1.52=.2476+1.582x_2$$

$$x_2=\frac{1.2724}{1.582}=.804$$

**10. Constant Heat Change.**—Another expansion of great importance in steam engineering is a change in pressure without any change in heat contents. This, obviously, is an adiabatic change but it is an irreversible process and its entropy is not constant as in the reversible adiabatic just studied. No mechanical work is done during this change.

The two most common occurrences of this expansion are when steam is reduced in pressure by a reducing valve and in the "wiredrawing" through the valves and passages of an engine or turbine. In both these cases the reduction in pressure is accompanied by an increase in velocity of the steam, but as the steam is brought more or less to rest after the reduction in pressure the heat given up to cause the velocity is returned to the steam by impact and friction.

This change can best be studied by means of the Mollier diagram. Suppose steam at 200 lbs. absolute pressure at 98 per cent quality is reduced in pressure through a reducing valve to a pressure of 35 lbs. absolute. Turning to the Mollier diagram we find at the intersection of the 200 lbs. line and the .98 quality line the total heat contents of one pound of steam =1,180 b.t.u. and the total entropy=1.525. As this is a reduction in pressure (expansion) without any change in heat contents, we follow across the heat contents line of 1,180 b.t.u. until it crosses the line of 35 lbs. pressure (interpolate between 30 and 40 lbs.). The quality is now 30° superheat—in other words the wet steam has become superheated during the reduction in pressure. This should naturally follow for a

constant heat change, as some of the heat of the liquid ($q$) at 200 lbs. has now been converted into superheat at the reduced pressure of 35 lbs. where $q_2$ is smaller. (See values of $q_1$ $q_2$ $r_1$ and $r_2$ in tables.) During this reduction in pressure the entropy has increased from 1.525 to 1.707. While this expansion is adiabatic it is not isentropic.

This reduction in pressure without any loss in heat contents is the principle on which Prof. Peabody's throttling calorimeter for measuring the quality of steam, is based. Wet steam at a known pressure and unknown quality is expanded through a partly closed valve until it is superheated. The temperature and pressure after throttling of the steam can be read and its heat contents computed. The heat contents before and after throttling are of course the same, and $x_1$ can therefore be found.

$$q_1+r_1x_1=q_2+r_2+Ct_s.$$

**11. The Rankine Cycle with Complete Expansion** (Clausius).—This cycle is generally spoken of as the cycle of the ideal engine

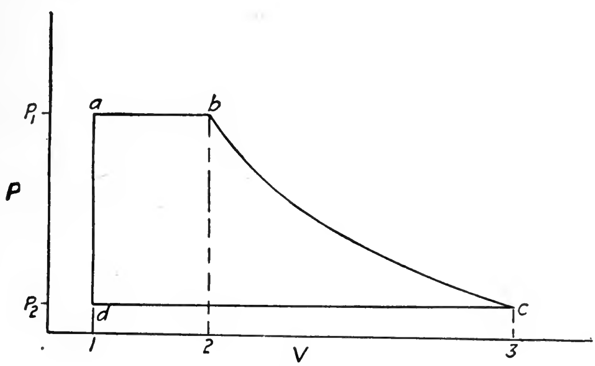


FIG. 5.

or more commonly as the Rankine cycle. Some writers refer to it as the Clausius cycle reserving the word Rankine to the cycle with incomplete expansion. In this book it will be referred to as the Rankine cycle. The pressure volume ($PV$) diagram of this theoretical cycle is represented in Fig. 5 and the $T\phi$ diagram in Fig. 6. It consists of a constant pressure ($P_1$) admission from the boiler along $ab$; cut off from the boiler at $b$; adiabatic expansion from $b$ to $c$ at back pressure

$P_2$; exhaust to the condenser along *cd;* and constant volume compression along *da* to the initial pressure $P_1$.

The cycle should not be thought of as taking place in an engine cylinder but in a plant, consisting of boiler, engine, condenser and necessary pumps—in other words it is in its broadest sense a plant cycle.

By referring to the *PV* and $T\phi$ diagrams the following discussion is readily followed: At (*a*) the working substance, considered for convenience as one pound of water, is at the boiling point (heat contents $q_1$). As heat is applied an isothermal expansion takes place at constant pressure and the water is converted into dry steam. The heat received by the water in changing from water at (*a*) to dry steam at (*b*) ($T\phi$ diagram) is (*a-b-3-2*) or $r_1$. The steam passes to the engine as it is formed, moves the piston from *a* to *b* (*PV* diagram) and does the external work $Ap_1u_1$ b.t.u. (*a-b-2-1* on *PV* diagram).

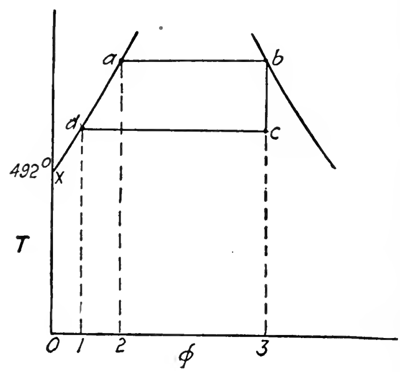


FIG. 6.

The steam now expands adiabatically from *b* to *c* (pressure $P_2$) doing the work $E_1 - E_2 = (q_1 + \rho_1) - (q_2 + x_2\rho_2)$ at the expense of the internal energy in the steam. (Area *b-c-3-2* on *PV* diagram.) No heat is received or given up and no area is included under the line *bc* on the $T\phi$ diagram.

The piston is now moved from *c* to *d* which rejects the working substance (steam) to the condenser. The heat of vaporization $x_2r_2$ is now carried away by the cooling water (area *c-3-1-d* on $T\phi$ diagram). The work done on the piston (*PV* diagram) is *c-3-1-d* ($Ap_2x_2u_2$ b.t.u.). This is an isothermal compression. The condensed water with heat contents $q_2$ (area *o-x-d-1* on $T\phi$ diagram) is now returned to the boiler (as feed water) and is heated again to the boiling point at point (*a*). The heat added to the water ($T\phi$ diagram) is area *d-a-2-1* which is $q_1 - q_2$. Neglecting the very small volume change in heating the water from (*d*) to (*a*), this is a constant volume change (*PV* diagram) and no work is done.

Collecting our terms, we have the work done during the cycle from the $PV$ diagram is $W = (a\text{-}b\text{-}2\text{-}1) + (b\text{-}c\text{-}3\text{-}2) - (c\text{-}3\text{-}1\text{-}d) =$ area *abcd*. From the thermodynamic standpoint we have:

$$\begin{aligned} W &= Ap_1u_1 + (q_1 + \rho_1 - q_2 - x_2\rho_2) - Ap_2x_2u_2 \\ &= q_1 + r_1 - q_2 - x_2r_2 \\ &= H_1 - H_2 \end{aligned}$$

where

$$Ap_1u_1 + \rho_1 = r_1$$

and

$$Ap_2x_2u_2 + x_2\rho_2 = x_2r_2$$

$u_1$ = the increase in volume during vaporization.

$x_2u_2$ = decrease in volume during condensation.

From the $T\phi$ diagram the heat received is

$$Q_1 = (a\text{-}b\text{-}3\text{-}2) + (d\text{-}a\text{-}2\text{-}1) = r_1 + q_1 - q_2 = H_1 - q_2$$

and the heat rejected is

$$Q_2 = (c\text{-}3\text{-}1\text{-}d) = x_2r_2$$

The work done becomes

$$Q_1 - Q_2 = r_1 + q_1 - q_2 - x_2r_2 = H_1 - H_2$$

This is the same expression as found from the $PV$ diagram. The efficiency of the cycle becomes

$$\text{Eff.} = \frac{\text{output}}{\text{input}} = \frac{\text{work done}}{\text{heat received}} = \frac{H_1 - H_2}{H_1 - q_2}$$

In this book,

$Q_1$ = heat received.
$Q_2$ = heat rejected.
$\lambda$ = total heat of steam from tables.
$H_1$ = heat contents for condition 1 $(= x_1r_1 + q_1)$
$H_2$ = heat contents for condition 2 $(= x_2r_2 + q_2)$

The above method of explaining the Rankine cycle is not the one used in treatises on thermodynamics but is a most instructive one for the student who has been through a regular course in thermodynamics and is familiar with the customary presentation of cycles.

This is the standard cycle used as a basis of comparison for all types of steam engines and turbines and the above efficiency will be used constantly in this book.

The work done in the above cycle per pound of steam is $H_1 - H_2 = (H_1 - q_2) - x_2r_2$ which is the total heat contents for condition $P_1$ minus the total heat contents after an adiabatic expansion. The heat received is

$$H_1 - q_2 = Q_1$$

and the heat rejected is,

$$x_2r_2 = Q_2$$

It should be noted that $H_1$ is not the heat received and $H_2$ the heat rejected; the heat received $(H_1 - q_2)$ minus the heat rejected $(x_2r_2)$ comes out equal to the difference of the initial and final heat contents $(H_1 - H_2)$. Again, while the initial heat contents often represents the total heat of the steam read directly from the steam tables at pressure $P_1$, the final heat contents $H_2$ does not represent the total heat contents at pressure $P_2$ as given in the steam tables, but $H_2 = q_2 + x_2r_2$, where $x_2$ is the quality after an adiabatic expansion.

The work done and the efficiency of the Rankine cycle can very quickly be obtained by the use of the Mollier diagram. As for example, suppose an engine is operating between an initial pressure of 200 lbs. abs. dry steam and a back pressure of 4 lbs. absolute.

From the Mollier diagram

$$H_1 = 1{,}197 \text{ b.t.u.}$$

$$\phi_1 = 1.545$$

Now by going down the constant entropy line $\phi = 1.545$ we have

$$H_2 \text{ at 4 lbs. } abs = 933 \text{ b.t.u.}$$

$H_1 - H_2$ now becomes $1{,}197 - 933 = 264$ b.t.u., and by looking up $q_2$ in the steam tables at 4 lbs. abs.

$$q_2 = 121 \text{ b.t.u.}$$

$$H_1 - q_2 = 1{,}076 \text{ b.t.u.}$$

$$\text{Rankine Eff.} = \frac{H_1 - H_2}{H_1 - q_2} = \frac{264}{1{,}076} = 0.245$$

**12. Rankine Cycle with Superheated Steam.**—If superheated steam is used instead of wet or saturated steam the heat received becomes

$$H_1 = q_1 + r_1 + Ct_s$$

The $T\phi$ diagram is shown in Fig. 7. The area *abcdgh* represents the heat received, the area *bcd* represents the superheat ($Ct_s$). The remainder of the cycle and the $T\phi$ diagram is the same as for saturated steam.

**13. Steam Consumption of Rankine Engine.**—In the Rankine cycle we have found that each pound of steam does the work equivalent of $H_1-H_2$ b.t.u. As it takes 2,545 b.t.u. per hour to produce one horsepower (33,000 ft. lbs. per min.), the theoretical steam consumption for an ideal engine working on the cycle becomes

$$W_t = \frac{2{,}545}{H_1-H_2} \text{ lbs. per hp. per hour.}$$

The above expression for the steam consumption of a theoretical engine working on the Rankine cycle is used as a basis for estimating the steam consumption of engines and turbines and comparing the performance of propelling machinery. It will be referred to again in a later article.

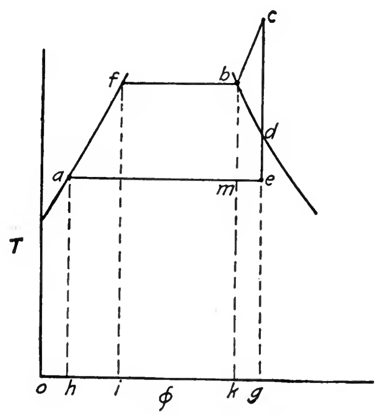

FIG. 7.

**14. The Actual Engine.** — The cycle of the Rankine engine is a theoretical one that cannot be realized in practice. If we let *abcd* (Fig. 8) represent the *PV* diagram of the Rankine cycle, *1–2–3–4–5* will represent the work done by a real engine working between the same initial pressure and the same back pressure. We will now review briefly some of the causes for the difference between the real and theoretical engine.

In place of the theoretical engine with non-conducting walls we now have an engine in which the walls absorb and radiate heat; instead of the engine being lifted from the hot body and to the cold body or connected directly to boiler and condenser we have long pipe lines between the boiler and engine, and engine and condenser. The steam which was at pressure $P_a$ at the boiler has now dropped to $P_1$ due to radiation, condensation and wiredrawing through the pipe and valves. Instead of the steam entering the cylinder along a constant pressure line *ab* it enters the cylinder along *1–2*, the pressure and volume being reduced to condition *2* at cutoff. The reduction in pressure is due largely to wiredrawing through the valves and valve passages.

As soon as the hot steam enters the cylinder it strikes the relative cool

walls, which have just been in contact with the exhaust steam at a low temperature, and gives up some of its heat to the walls. This causes condensation of the steam to take place, known as "initial condensation." This condensation continues through admission and accounts for the point of cutoff in the real engine being brought in to *2*. This initial condensation is one of the largest losses in a steam engine and amounts from anywhere from 10 to 40 per cent in single cylinder engines.

After cutoff, condensation continues and the quality falls off at a faster rate than due to adiabatic expansion until the temperature of the expanding steam has been reduced to the temperature of the walls. With a still further reduction in the steam temperature due to expansion, the temperature of the steam becomes lower than that of the walls and the flow of heat is from the walls to the steam. This causes the heat to re-evaporate from the walls and the quality to increase.

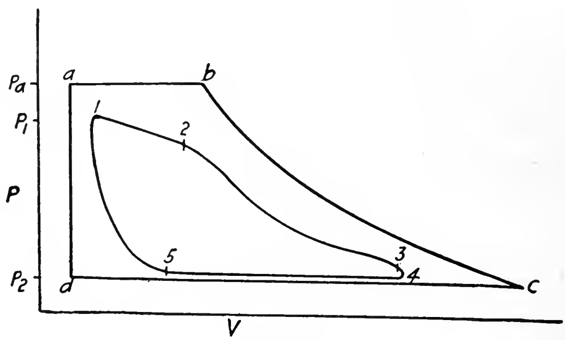


FIG. 8.

For mechanical reasons and to reduce the cost and size of the engine, the expansion is not carried down to the back pressure but is stopped at a higher pressure $P_3$. The exhaust valve now opens and release takes place from 3 to 4. During the pressure drop from 3 to 4 no work is done and the heat contents at 4 are practically the same as at 3. From 4 to 5 the piston returns and the remaining steam is expelled from the cylinder. There is a slight rise in pressure from 4 to 5 caused by the resistance of the exhaust passages.

During exhaust the walls are at a higher temperature than the steam and this causes some of the moisture on the walls to evaporate and heat passes from the cylinder walls to the steam. Thus some of the useful heat that was given up by the entering steam to the walls is now returned to the steam to be carried out and rejected in the condenser.

At 5 the exhaust valve closes and the steam is compressed to 1. This compression is necessary to bring the piston to rest without shock before starting on the new stroke. A clearance volume must be left at the end of the cylinder to allow space for the cushion steam at compression. This causes the difference in volume of point *a* and *1*.

The above very briefly describes the reasons for the differences between the ideal and real engine. It is intended only as a review of the students' more complete study of the subject in thermodynamics.

**15. Heat Losses in Actual Engine.**—The main heat losses in the cylinder, i.e., wiredrawing, condensation, incomplete expansion, radiation and clearance volume, have been treated briefly in the preceding paragraph. The largest loss in operating a heat engine is the heat rejected in the exhaust $x_2r_2$ which cannot be recovered or used. In order to reduce this rejected heat to as small amount as possible, condensers are used to reduce the back pressure and cause more of the heat to be converted into work. This is clearly seen by studying the $T\phi$ diagram.

**16. Thermal Efficiency.**—The thermal efficiency of a steam engine is,

$$\text{T.E.} = \frac{\text{output in b.t.u. per hour}}{\text{input in b.t.u. per hour}}$$

Reducing the above to the basis of one horsepower we have,

$$\text{T.E.} = \frac{2{,}545}{W_a(H_1 - q_2)}$$

where

$2{,}545$ = the heat equivalent of one horsepower per hour
$W_a$ = the lbs. of steam used per horsepower per hour
$H_1 - q_2$ = the input per lb. of steam in b.t.u.'s.

The b.t.u. supplied per pound is not $H_1$ but $H_1 - q_2$, as the heat of liquid ($q_2$) at exhaust pressure is not thrown away but is retained in the condensate and returned to the boiler, and hence should not be charged up against the engine as a loss.

**17. Efficiency Ratio.**—This is the ratio of the thermal efficiency of the actual engine to the efficiency of the Rankine engine. It is the universal accepted standard for measuring and comparing the performance of an engine and is the official standard for the Institution of Civil Engineers and the American Society of Mechanical Engineers.

$$\text{E.R.} = \frac{\dfrac{2{,}545}{W_a(H_1 - q_2)}}{\dfrac{H_1 - H_2}{H_1 - q_2}} = \frac{2{,}545}{W_a(H_1 - H_2)}$$

It will be observed that the above expression can be reduced to,

$$\text{E.R.} = \frac{W_t}{W_a}$$

Where $W_t = \dfrac{2{,}545}{H_1 - H_2}$ = steam consumption of the Rankine engine in pounds per horsepower per hour. This gives us the very simple expression $(W_t/W_a)$, the ratio of the theoretical to actual steam consumption, as the efficiency ratio.

*Example.*—S.S. "Rampo" full power trial $P_1$ (high pressure chest) 220 lb. gauge (235 lbs.).

Vacuum in condenser = 27.9 in.
Steam consumption of main engines = 12.8 lbs. per I.H.P. per hour.
Barometer = 30.48 in.

$P_2 = 2.58$ in. $Hg = 1.27$ lb./sq. in. (110°F.)
$H_1 = 1{,}200.3$ b.t.u., $\phi_1 = 1.533$ (Mollier diagram)
$H_2 = 867$ b.t.u. (Mollier diagram)
$q_2$ at 1.27 lb./sq. in. $= 78$ b.t.u.

By using entropy equation

$$\theta_1 + r_1/T_1 = \theta_2 + x_2 r_2/T_2$$

we have

$$.5603 + .9725 = .1473 + 1.8088x_2$$
$$x_2 = .765$$
$$H_2 = q_2 + x_2 r_2 = 78 + .765 \times 1{,}030 = 867 \text{ b.t.u., as before.}$$
$$H_1 - H_2 = 333 \text{ b.t.u.}$$
$$H_1 - q_2 = 1{,}122 \text{ b.t.u.}$$
$$W_t = \frac{2{,}545}{H_1 - H_2} = \frac{2{,}545}{333} = 7.64 \text{ lbs. per I.H.P. per hour.}$$
$$W_a = 12.8 \text{ lbs. per I.H.P. per hour.}$$
$$\text{Thermal efficiency (T.E.)} = \frac{2{,}545}{W_a(H_1 - q_2)} = \frac{2{,}545}{12.8 \times 1122} = .177$$
$$\text{Rankine Eff.} = \frac{H_1 - H_2}{H_1 - q_2} = \frac{333}{1122} = .297$$
$$\text{Eff. Ratio.} = \frac{7.64}{12.8} = .597$$
$$= \frac{.177}{.297} = .597$$

(Peabody's Steam Tables were used in the above calculations.)

**18. Shaft Horsepower and Indicated Horsepower.**—It is customary to express the power of reciprocating engines in indicated horsepower (I.H.P.) and turbines in shaft horsepower (S.H.P.). The power of Diesel engines and other types of internal combustion engines is generally expressed in brake horsepower (B.H.P.) which is the same as the S.H.P.

In the case of reciprocating engines the indicated horsepower has been used because of the early adoption of the steam engine indicator for measuring the power of an engine. This has continued as the basis because of the ease with which the I.H.P. can be measured; and because of the difficulty and unsatisfactory results due to the uneven turning moment, the S.H.P. or B.H.P. is seldom measured. The indicated horsepower is thus the power developed in the engine cylinders and is greater than that delivered to the propeller shaft because of the losses due to engine friction.

With turbines, on the other hand, the internal horsepower developed by the steam within the turbine cannot be measured; hence the S.H.P. delivered to the shaft has been used as the basis. The S.H.P. is readily measured by means of a torsion meter.

The steam consumption is always expressed in pounds per indicated horsepower per hour for reciprocating engines and in pounds per shaft horsepower per hour for turbines. This different basis of stating the steam consumption should cause no confusion, however, in working out efficiency ratios for engines and turbines. In comparing horsepowers and steam consumptions the different bases should be kept in mind. The efficiency ratios of a turbine will run somewhat lower and unit steam consumptions somewhat higher than a reciprocating engine which delivers the same power to the propeller with the same total steam consumption.

*Example.*—Suppose two sister ships require 5,000 horsepower to be delivered to the propeller. One ship is fitted with a reciprocating engine and the other with a turbine and each uses 65,000 lbs. of steam per hour at full speed. The S.H.P of the turbine is 5,000 and the I.H.P. of the engine is 5,420.

$$W_a \text{ (turbine)} = \frac{65{,}000}{5{,}000} = 13 \text{ lbs. per S.H.P. per hour.}$$

$$W_a \text{ (engine)} = \frac{65{,}000}{5{,}400} = 12 \text{ lbs. per I.H.P. per hour.}$$

Assuming $W_t = 7.0$ lb. per hp. per hour.
Eff. Ratio (turbine) $= .538$
Eff. Ratio (engine) $= .582$

The above is strictly a theoretical example for purpose of illustration. In an actual case the performance of the turbine of the first ship would be better than the reciprocating engine in the second ship, resulting in a lower hourly steam consumption and a lower steam consumption per S.H.P. per hour and a higher efficiency ratio.

**19. Mechanical Efficiency.**—This is the ratio of the shaft horsepower(S.H.P.) or brake horsepower(B.H.P.) to the indicated horsepower (I.H.P.).

$$\text{Mech. Eff.} = \frac{\text{S.H.P.}}{\text{I.H.P.}} = \frac{\text{B.H.P.}}{\text{I.H.P.}}$$

*Example.*—The mechanical efficiency of the reciprocating engine mentioned in the preceding article is,

$$\text{Mech. Eff.} = \frac{5{,}000}{5{,}420} = 0.92$$

The efficiency of the mechanism is 92 per cent, or in other words the mechanical losses between the cylinder and the shaft are 8 per cent. The mechanical efficiencies of marine engines range between 85 and 95 per cent; the common practice in absence of definite data is to assume the mechanical efficiency equal to 92 per cent when fitted with Kingsbury thrust blocks and engine driven air pump. The mechanical efficiency of the naval type of reciprocating engine with forced lubrication and independent pumps is, of course, higher than the merchant type with attached air pump. The mechanical efficiency of the Diesel engine is much lower than the steam engine because of the large number of pumps, air compressors, etc., driven by the engine. The use of Kingsbury thrust blocks in place of the horseshoe type increases the mechanical efficiency about 5 per cent.

**20. Heat Consumption.**—The true basis for stating the performance of an engine is the b.t.u. consumption instead of the steam consumption. Because of the differences in initial pressures and vacuum that may exist for different engines, a comparison of the steam used per horsepower per hour is not a true measure of the relative efficiencies of the engines. However, if the consumption is expressed in b.t.u. consumed per horsepower per min., all differences in working pressures are eliminated. The b.t.u. supplied per pound of steam is $H_1 - q_2$, where $H_1$ ($= x_1r_1 + q_1$ or $r_1 + q_1 + ct_s$) is the heat contents of the steam under initial conditions and $q_2$ is the heat of the liquid at condenser pressure. As pointed out before, the heat $q_2$ is returned to the boiler and should not be charged against the engine. This value for $q_2$ should always be taken for the back pressure of the engine regardless of the temperature of the condensate leaving the condenser; also the same expression $(H_1 - q_2)$ is used for a

non-condensing engine where all the heat $(H_2 = x_2r_2 + q_2)$ is thrown away. The fact that this heat $q_2$ can be used again is all that we are interested to know in studying the performance of the engine itself. The b.t.u. consumption per horsepower per minute becomes,

$$= \frac{W_a(H_1 - q_2)}{60}$$

If an engine is supplied with jacket steam or reheater steam either at full boiler pressure or at reduced pressure, the heat consumption for these purposes must be added to that used in the cylinders. The heat of the liquid $q_2$ in the drips from the jackets will be that corresponding to the pressure of the steam in the jackets.

*Example.*—Consider an engine operating under the following conditions:

I.H.P. = 4,000
$P_1$ = 250 lbs. gage (dry steam)
$P_2$ = 7.97 in. *Hg* (3.9 lbs./sq. in.)
$W_a$ (cylinders only) = 12.0 lbs./I.H.P.
Jacket steam at 250 lbs. = 400 lbs./hour (0.10 lb./I.H.P. per hour)
$H_1 - q_2 = 1{,}202 - 120 = 1{,}082$ b.t.u.

$$\text{B.t.u. consumption} = \frac{(1{,}202-120)\times 12.0}{60} + \frac{(1{,}202-380)\times 0.10}{60}$$

$$= 216.2 + 1.4 = 217.4 \text{ b.t.u. per I.H.P. per minute.}$$

**21. The Cycle of the Direct Acting Steam Pump.**—This cycle consists of a constant pressure admission and a constant volume rejection connected by two constant volume lines—in other words the *PV* diagram is rectangular (Fig. 9). Steam is admitted at pressure $P_1$ during the whole stroke from *a* to *b* and then released at *b* under full boiler pressure to the exhaust line. At release the pressure drops to the back pressure $P_2$ almost instantly, giving a constant volume line *bc* on the *PV* diagram, i.e., there is a pressure drop from *bc* without any movement of the piston. The volume of steam after leaving the cylinder naturally increases at the reduced pressure. The piston now returns (*cd*) and there is practically no compression at the end of the stroke. The exhaust valve closes at *d*, the inlet opens at the same time and the pressure increases from *d* to *a* at constant volume.

The heat received by the pump per pound of steam is,

$$Q_1 = x_1r_1 + q_1$$

The work done $W = P_1(v_b - v_a) - {}_2P(v_c - v_d) = P_1x_1u_1 - P_2x_2u_2$ ft. lbs. (*PV* diagram)

$$W = (AP_1x_1u_1 - AP_2x_2u_2) \text{ b.t.u.}$$

The heat rejected by engine in the exhaust is $Q_2 = Q_1 - W$

$$Q_2 = x_1r_1 + q_1 - AP_1x_1u_1 + AP_2x_2u_2$$
$$Q_2 = (x_1r_1 + q_1 - AP_1x_1u_1) + AP_2u_2$$
$$Q_2 = (x_1\rho_1 + q_1 + AP_2x_2u_2)$$

The heat $x_1\rho_1+q_1+AP_2x_2u_2$ is rejected to the condenser or is available for feed heating. At release ($b$) there is a sort of free expansion from $P_1$ to $P_2$; no work is done and consequently no heat is lost. The heat in the exhaust steam is therefore that in the steam at release. At first thought the conclusion might be drawn from the $PV$ diagram that the heat in the steam delivered to the exhaust line would be the total heat of the steam at full boiler pressure $P_1$. This, however, cannot be the case as the steam has already done some mechanical work in moving the piston from $a$ to $b$. The heat required for this work is $AP_1x_1u_1-AP_2x_2u_2$. This leaves the heat $(x_1\rho_1+q_1+A\rho_2x_2u_2)$ to be discharged to the exhaust line at pressure $P_2$.

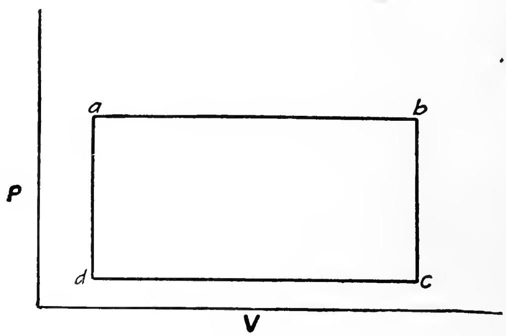


FIG. 9.

During the free expansion from $b$ to $c$ the steam if initially dry now becomes superheated at pressure $P_2$. An example will illustrate the above:

$$P_1=200 \text{ lbs. abs.}$$
$$x_1=100 \text{ per cent}$$
$$P_2=20 \text{ lbs. abs.}$$

Steam used per hour = 100 lbs.
Heat in the dry steam when released at 200 lbs. is,

$$(\rho_1+q_1)\times 100=111{,}310 \text{ b.t.u.}$$

Heat in 100 lbs. of exhaust steam when reduced to 20 lbs. abs. is,

$$111{,}310+AP_2u_2=118{,}740 \text{ b.t.u.}$$

Heat in 100 lbs. dry steam at 20 lbs. abs. is,

$$(r_2+q_2)\times 100=115{,}580 \text{ b.t.u.}$$

Increase in contents $=118{,}740-115{,}580=3{,}160$ b.t.u.

Degree of superheat $=\dfrac{31.60}{0.46}=68.5°$ approx.

A small decrease in the quality of the steam at release would materially lower the degree of superheat at exhaust and might reduce it to wet steam.

In the foregoing discussion the terms used for $Q_1$ and $Q_2$, the heat received and rejected by the engine cylinder, should be distinguished from $(H_1-q_2)$ the heat received and $x_2r_2$ the heat rejected, when considering a plant cycle.

The above discussion has been presented for its value in connection with a study of feed heating by exhaust steam. The heat in the exhaust available for feed heating is $x_1\rho_1+q_1+Ap_2x_2u_2$ which includes $q_2$. A large number of the auxiliaries used on shipboard are driven by direct acting pumps. The above discussion shows that even with wet steam at release the quality of steam rejected to the auxiliary exhaust line by this type of pump is practically dry and may perhaps be superheated.

# CHAPTER III

## FUELS

**22. General.**—Fuels used on shipboard are almost entirely limited to two kinds—bituminous and semi-bituminous coal, and fuel oil. Anthracite coal and gasoline are used to a very small extent. Other fuels such as wood, coke, gaseous fuel, and powdered fuels used to a considerable extent on shore find practically no use at sea.

A comparison of the merits of coal and oil as a fuel for use on shipboard is taken up in Chap. VI after a study has been made of boilers and combustion.

**23. Composition and Classification of Coal.**—Coal is classified in various manners according to its physical or chemical characteristics. The common method is to classify it according to the percentage of volatile matter or hydrocarbons which it contains. There are no definite boundaries between the various groups. The following table presents a general grouping of coal by this method:

| Kind of Coal | Per Cent Volatile Matter |
|---|---|
| Anthracite | 3– 8 |
| Semi-anthracite | 8–12 |
| Semi-bituminous | 15–25 |
| Bituminous (Eastern) | 25–40 |
| Bituminous (Western) | 35–50 |
| Lignite | 50 and over |

Chemically, coal consists mainly of Carbon, Hydrogen, Oxygen, Nitrogen, Sulphur, ash and moisture. The exact nature in which these elements are combined is not clearly known. The analysis of coal into the elements mentioned above but without regard to their chemical combinations is known as the *ultimate analysis*.

**24. Proximate Analysis.**—Another form of analysis known as the *proximate analysis* is also used. This is a much simpler analysis to make and is of great importance to the engineer.

In this analysis the percentage of moisture, volatile matter, fixed carbon and ash is determined. The hydrocarbons and other gaseous constituents of the coal that distill off upon heating are termed *volatile matter;* the uncombined carbon is termed *fixed carbon* and the residue left after burning the volatile matter and fixed carbon is the ash.

With proper facilities the proximate analysis is easily made. A sample of coal is weighed and then heated at a temperature of 221°F. for about an hour and a half. The material lost during this heating is termed *moisture.* This definition of moisture is of course arbitrary, as all of the moisture may not be driven off and on the other hand some of the lighter volatile matters are often distilled off at a temperature of 221°F. The moisture percentage naturally varies greatly depending on atmospheric and storage conditions. The residue after driving off the moisture is termed *dry coal.* After driving off the moisture the coal is subjected to a very high temperature in a covered vessel and maintained at this temperature for about 7 minutes. The material driven off is termed the *volatile matter* and that remaining after subtracting out the ash is termed *fixed carbon.* The fixed carbon is practically the same as coke. The fixed carbon plus the volatile matter is termed the *combustible.*[1]

Table II gives the ultimate analysis and proximate analysis and heat of combustion of a few representative coals. The reader is referred to either Kent's Pocket-book or Mark's Handbook and the Bulletins of the U. S. Geological Survey for a more complete list. As the proximate and ultimate analyses are made both on "coal as received" and "dry coal," care must be used in comparing analyses to make certain which basis is used.

**25. Calorific Value of Coal.**—The calorific value or heat of combustion of coal is the number of b.t.u. that are given up in burning one pound of coal. The heat value is expressed in b.t.u. per lb. of coal "as received," per lb. of "dry coal" and per lb. of "combustible." The calorific value of coal varies roughly between 13,500 and 15,800 b.t.u. per pound. The highest values (15,800) are for semi-bituminous coal where the percentage of fixed carbon is between 75 and 85 per cent. The anthracites with higher percentages of fixed carbon and the bituminous coals with lower percentages have lower calorific values.

[1] See *U. S. Bureau of Mines Bull.* **41.**

TABLE II.—REPRESENTATIVE COALS

| Kind of coal | Location | Proximate analysis | | | | Ultimate analysis | | | | | | Heat of combustion dry coal b.t.u per lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Moisture | Volatile matter | Fired carbon | Ash | C | H | N | O | Ash | S | |
| Anthracite | Lehigh | 2.51 | 5.75 | 84.72 | 7.02 | 87.44 | 1.97 | 0.77 | 2.03 | 7.20 | 0.59 | 13,730 |
| Semi-bituminous | Pocahontas | 0.65 | 17.13 | 76.06 | 6.16 | 82.72 | 4.63 | 1.31 | 4.42 | 6.19 | 0.72 | 14,817 |
| Semi-bituminous | George's Creek | 4.03 | 18.79 | 69.24 | 7.94 | 79.19 | 4.85 | 2.12 | 4.63 | 8.27 | 0.94 | 14,386 |
| Bituminous | Pittsburgh, Steaming | 1.92 | 36.63 | 54.24 | 7.21 | 76.59 | 5.21 | 1.59 | 7.70 | 7.35 | 1.56 | 13,889 |
| Bituminous | Hocking, Ohio | 6.13 | 35.35 | 50.93 | 7.59 | 73.12 | 4.94 | 1.51 | 10.96 | 8.06 | 1.41 | 12,633 |
| Brown lignite | North Dakota | 12.89 | 46.96 | 30.41 | 9.74 | 58.33 | 4.33 | 0.81 | 23.40 | 11.18 | 1.95 | 9,657 |

Semi-bituminous coal is found in the United States in the Appalachian Range in Pennsylvania and West Virginia. It is a peculiar fact that the heat value of coal found in the United States decreases roughly as the distance of the mine from these semi-bituminous mines mentioned above.

The calorific value of coal can be found approximately from the ultimate analysis by Dulong's formula:

$$\text{B.t.u. per lb.} = 14{,}600C + 62{,}000(H - O/8) + 4{,}000S.$$

For the ordinary run of coals used on shipboard this formula will give values within 1 1/2 per cent or 200 b.t.u. Unless care is taken to obtain a good representative sample the b.t.u. value found by means of the bomb calorimeter is not good within 100 b.t.u.

**26. Anthracite.**—Anthracite coal is a hard coal consisting of about 88 per cent fixed carbon and 3 to 6 per cent volatile matter. It requires nearly twice as much grate surface as bituminous coal, is a slow burning coal with a short flame, and burns with practically no smoke. For the smaller sizes a heavy draft is required as shown by Fig. 38. Anthracite finds practically no use on shipboard largely because of its high cost and slow burning qualities.

**27. Semi-Bituminous.**—This grade of coal is intermediate between anthracite and bituminous. The fixed carbon runs around 75 per cent and the volatile matter around 18 per cent. Semi-bituminous coal is very high in heat value, and low in ash and moisture content. It burns with a comparatively short flame and is an excellent burning and coking coal. Because of its low percentage of volatile matter it burns with much less smoke than bituminous coal. This coal is the most desirable of all for marine purposes and is used exclusively in the U. S. and British Navies for bunker coal.

**27a. Bituminous Coal.**—This grade of coal is the most extensively distributed of all the coals. It furnishes by far the largest percentage of steaming and bunker coal. Bituminous coals are distinguished by the very high percentage of volatile matter they contain and hence give forth large volumes of smoke unless carefully fired. A large variety of grades is included under the name of bituminous and include *dry* and *caking* coals, *long flame* and *short flame* coals. The dry bituminous coals

are much to be preferred for use under boilers as they burn freely without fusing and with a short flame. The caking coals on the other hand swell up and fuse together in the furnace and burn with a long flame. Bituminous coal is used extensively in the merchant service.

**28. Powdered Coal.**—Powdered coal, consisting of the dust and mine culm and also the pulverized cheaper and poorer grades of coal, is being used to a large extent at the present time in power plants on shore and may eventually find use on shipboard. This is dealt with further in Chap. VII.

**29. Purchase and Selection of Coal.**—Specifications should be always laid down before purchasing coal for bunker use. These specifications are sometimes very rigid and sometimes very loose. Often the name of the mine from which the coal was mined is sufficient guarantee as to its quality. The following characteristics should be ascertained before purchasing:

1. Ash and moisture content.
2. Burning characteristics.
3. Calorific value.
4. Size and purity of coal (percentage of lumps, slate, etc.).

The following requirements that must be met by semi-bituminous coal for the U. S. Navy are excellent and should serve as a guide in purchasing coal for the merchant service:

1. Moisture in "delivered coal" not over 3 per cent.
2. Ash in "dry coal" not over 7 1/2 per cent.
3. Volatile matter not over 20 per cent.
4. Sulphur in "dry coal" not over 1.25 per cent.
5. B.t.u. per lb. "dry coal" at least 14,700.
6. Lump, at least 40 per cent.

The necessity of low moisture and ash content of the coal is more important on shipboard than for stationary plants, for here the added weight and space taken up by the moisture and ash reduce the cargo carrying capacity of the ship and increase the labor of handling in the fire room. Moisture and ash also cause losses in combustion that will be treated in Chap. V.

The percentage of moisture will vary greatly, depending upon the handling and exposure of the coal after leaving the mine. Some limit of moisture should always be insisted upon, however. Some idea of the burning qualities of coal can be obtained from the analysis, especially from the percentage of volatile matter

and sulphur. The former will given an indication of the burning qualities and the latter an indication of the probability of the fusing and clinkering of the ash. Experience with the burning characteristics of coals from the various regions is, however, fairly essential in the selection of a suitable marine coal. The U. S. Navy keeps an "acceptable list" of mines from which they have found the coal to be of good quality, suitable, and satisfactory. In comparing prices of coals all should be reduced to a common basis of ash, moisture and heat content. Merchant ships, especially tramps, that are forced to purchase coal in the various ports of the world, cannot insist upon very exact specifications; yet some requirements should be insisted upon. Previous experience with a particular coal is often a good basis for selection but this gives very little indication of heat and moisture content.

The U. S. Bureau of Mines has drawn up standard specifications for use in purchasing and sampling coal. An extract of these is given in Mark's Handbook.

**30. Fuel Oil.**—Practically all of the fuel oil used today is the residue of petroleum or crude oil after the volatile oils, gasoline, kerosene and the lubricating oils have been distilled off. The residue is heavier, less volatile, cheaper, and safer after the more valuable oils mentioned above have been removed. · Crude oil is practically never used as a fuel because of the valuable distillates that it contains. Shale and tar oils are not used today for producing fuel oil to any extent but in the future they may be a valuable source for fuel. The petroleum yield of the United States is divided in the following approximate proportions:

| | Per Cent |
|---|---|
| Gasoline | 9 |
| Kerosene | 30 |
| Gas oil | 9 |
| Lubricating oil | 9 |
| Fuel oil | 36 |
| Paraffin and miscellaneous | 7 |
| | 100 |

Petroleum is a mineral oil consisting of hydrocarbons and is divided into three main groups depending upon its base or residue left after distillation. These groups are: (1) paraffin base, comprising the oils found in the Appalachians and Middle West;

TABLE III

| Kind of oil | Gravity °B. | Flash point °F. | B.t.u. per gallon | C | H | N | S | O | Viscosity of 70°F. (Engler) |
|---|---|---|---|---|---|---|---|---|---|
| Texas Oil Co. | 24.0 | 210 | 145,000 | 86.40 | 12.25 | 0.58 | 0.74 | 0.03 | From 5 to 10 |
| Texas Co., Mexican | 20.0 | 65 | 143,720 | 84.40 | 11.80 | 0.50 | 2.80 | 0.50 | (?) |
| California Oil | 16.3 | 240 | 157,740 | 86.93 | 11.66 | 0.75 | 0.48 | 0.18 | (?) |
| Texas Co., Mexican | 17.3 | 126 | 150,250 | 84.00 | 11.70 | 0.72 | 3.14 | 0.44 | 116.1 at 100°F. |
| Texas Co. oil | 23.6 | 207 | 147,600 | 86.20 | 12.70 | 0.25 | 0.50 | 0.35 | .5 |
| Standard Oil, Mexican | 17.1 | 135 | 150,250 | 83.50 | 11.35 | 0.42 | 3.06 | 1.67 | 300.5 |
| Standard Oil, Mexican | 15.4 | 202 | 151,390 | 83.90 | 11.55 | 0.42 | 3.47 | 0.66 | 715 |
| Illinois oil | 27.2 | 146 | 145,630 | 84.60 | 13.05 | 0.27 | 0.30 | 1.78 | 6.8 |
| Indiana oil | 29.6 | 144 | 144,250 | 84.20 | 13.65 | 0.15 | 0.50 | 1.50 | 4.83 |
| Louisiana oil | 19.8 | 275 | 147,160 | 86.30 | 12.30 | 0.14 | 0.43 | 0.83 | 50.2 |
| "Star" California oil | 23.9 | 180 | 146,070 | 86.40 | 12.20 | 0.33 | 0.42 | 0.65 | 2.96 |
| "Richmond," Cal. oil | 17.1 | 228 | 151,210 | 86.60 | 11.70 | 0.58 | 0.70 | 0.42 | 82.2 |
| Lima, Ohio | 30.4 | 149 | 142,660 | 85.20 | 12.80 | 0.12 | 0.40 | 1.48 | 7.0 |
| Mexican gas oil | 34.2 | 151 | 139,760 | 84.70 | 14.00 | 0.16 | 1.13 | 0.01 | 1.56 |
| Mexican residue | 10.0 | 374 | 153,910 | 83.60 | 12.00 | 0.45 | 3.86 | 0.09 | 67 at 200°F. |
| Gulf Ref. Co. oil | 29.2 | 170 | 144,015 | 85.00 | 13.50 | 0.10 | 0.20 | 1.20 | 10 |
| Union Oil Co., Cal. | 12.9 | 285 | 151,400 | 84.60 | 13.40 | 0.68 | 0.93 | 0.39 | 102 at 125°F. |
| Union Oil Co., Cal. | 13.2 | 262 | 152,465 | 85.00 | 12.50 | 0.77 | 0.83 | 0.90 | 80 at 125°F. |
| Union Oil Co., Cal. | 12.9 | 280 | 151,245 | 84.50 | 13.00 | 0.77 | 0.87 | 0.86 | 96 at 125°F. |
| Union Oil Co., residuum | 18.5 | 223 | 150,100 | 86.20 | 12.00 | 1.03 | 0.74 | 0.03 | 93.5 |
| Producers' crude, Union Oil Co. | 16.1 | 174 | 152,630 | 85.10 | 13.00 | 0.66 | 0.70 | 0.54 | 98.0 |
| Coalinga Field | 16.5 | 186 | 151,260 | 86.20 | 12.60 | 0.52 | 0.59 | 0.09 | 73.5 |
| Toltec oil | 11.7 | 220 | 149,330 | 83.80 | 11.20 | 0.43 | 4.50 | 0.07 | 166 at 125°F. |
| Avon, California | 17.1 | 168 | 148,560 | 85.50 | 12.10 | 0.66 | 0.78 | 0.96 | 65.2 |
| Gaviota, California | 17.1 | 230 | 147,745 | 83.00 | 11.50 | 0.59 | 2.51 | 2.40 | 89.2 at 100°F. |
| Anglo-Mexican Co. No. 1 | 15.8 | 188 | 148,500 | 83.80 | 11.75 | 0.37 | 3.18 | 0.90 | 47.6 at 100°F. |
| Anglo-Mexican Co. No. 2 | 16.2 | 238 | 150,500 | 84.00 | 12.00 | 0.36 | 2.78 | 0.86 | 71.54 at 100°F. |
| Anglo-Mexican Co. No. 3 | 17.3 | 164 | 148,850 | 83.40 | 11.95 | 0.34 | 3.40 | 0.91 | 96.38 at 100°F. |
| Camden Co. coal tar by-product | 6.0 | 108 | 158,120 | 90.90 | 8.00 | 0.14 | 0.46 | 0.50 | 2.3 |
| Mexican oil | 17.6 | 182 | 147,500 | 83.60 | 11.75 | 0.32 | 2.70 | 1.63 | 49.24 at 100°F. |
| Toltec or Panuco oil | 12.0 | 124 | 149,500 | 82.50 | 10.95 | 0.45 | 4.53 | 1.57 | 375 at 105°F. |

(2) asphaltic base, comprising the oils found in California and along the Gulf of Mexico; (3) the olefin base, comprising the Russian oils. Table III taken from the paper by J. J. Hyland (*Jour. A.S.N.E.*, May, 1914) gives the properties of the various American fuel oils.

**31. Properties of Fuel Oil.**—The following characteristics of fuel oils are usually determined before purchasing:

1. Specific gravity.
2. Flash point.
3. Calorific value.
4. Viscosity.
5. Water and sediment content.

An ultimate analysis is also sometimes made to determine the chemical properties of the oil.

The specific gravity of fuel oil is expressed in degrees Baumé at 60°F. referred to water at 60°F. as unity and is read by a hydrometer. The Baumé gravity can be converted into specific gravity by the following formula:

$$\text{Specific gravity} = \frac{140}{130 + {}^\circ\text{Baumé}}$$

The flash point is the temperature at which the volatile gases which cause an explosive mixture with air are given off. It is impossible to get an explosive mixture of air and the gases given off by the oil when heated to a temperature below the flash point. The *fire point* often used in connection with liquid fuels is the temperature at which the gases given off by the oil will burn continuously. This temperature is generally between 20° and 30° above the flash point.

The calorific value is expressed, the same as for coal, in b.t.u. per pound and also in b.t.u. per gallon. The b.t.u. per lb. varies between 18,000 and 19,000 and varies with the specific gravity. The following formula by Sherman & Kropff gives an approximate value:

$$\text{B.t.u. per pound} = 18{,}650 + 40({}^\circ\text{Baumé} - 10).$$

Viscosity, or the internal friction of oil, is measured by the rate of flow through an orifice at a given temperature. There are various scales used in which the amount of oil used, the temperature, and the size of the orifice are different; hence the instrument used should always be mentioned in expressing viscosity.

The Engler and Saybolt are the two most common viscosimeters used; the Engler is the standard for fuel oil in the U. S. Navy and the Saybolt is used pretty generally for lubricating oils. The viscosity in the Engler scale is found by dividing the time required for the flow of oil at 70°F. by the time required for the flow of an equal volume of water. Hence the viscosity of water is unity with this instrument. The Saybolt viscosity is the number of seconds required for the flow of 60 c.c. of liquid at 70°F. from the instrument. The viscosity of water by this scale is 30 seconds. The Engler reading can be approximately converted into Saybolt seconds by multiplying by 36. Thus $S = 36E$.

The barrel, consisting of 42 gallons, is the standard used in selling fuel oil. As oil is sold by volume and not weight a correction should always be made for temperature. A full comparison of coal and fuel oil as a fuel for steamships is given in Chap. VII; and colloidal fuel in Art. 77. For a complete discussion of fuel oil the reader should consult the two following excellent papers:

John J. Hyland. *Jour. Am. Soc. Naval Engineers*, May, 1914.
Ernest H. Peabody. Int. Engineering Congress, San Francisco, 1915.

# CHAPTER IV

## MARINE BOILERS

**32. Classification.**—Marine boilers in common use can be divided into four groups: (1) the cylindrical internally fired fire-tube, or Scotch boiler; (2) the large tube water-tube boiler; (3) the small tube water-tube boiler of light construction, generally known as the "express" type; and (4) the flash boiler with practically no water volume, in which the feed water is converted into steam almost immediately after entering the boiler.

The first two groups include practically all the boilers used on merchant ships, although a few recent passenger vessels have been fitted with light "express" type boilers. All large naval vessels such as battleships and cruisers are fitted with water-tube boilers of group (2). The "express" type boiler is used on all light high speed naval vessels such as destroyers and scouts where weights must be kept to a minimum. The flash type boiler has one advantage over all the other types because of its extremely light weight but has never been used except in very small units, mostly in launches.

**33. Scotch Boilers.**—The Scotch boiler has been used almost universally on merchant ships up to the opening of the world war. Today, while many of the older engineers still cling to this type, the large tube water-tube boiler (group 2) is replacing it to a large extent.

The construction of the Scotch boiler is clearly shown in the drawing, Fig. 10, and in Figs. 11, 12 and 13. These figures should be studied carefully in conjunction with the following description. It consists of a cylindrical plate shell and two flat plate ends which are flanged and riveted to the cylindrical shell. The furnaces, which are of corrugated steel, are fitted internally and hence entirely surrounded by water. Each boiler is fitted with two or more furnaces.

The furnaces are attached to combustion chambers located in

the back of the boiler; and the boiler tubes lead from the upper part of the combustion chamber and extend through the boiler

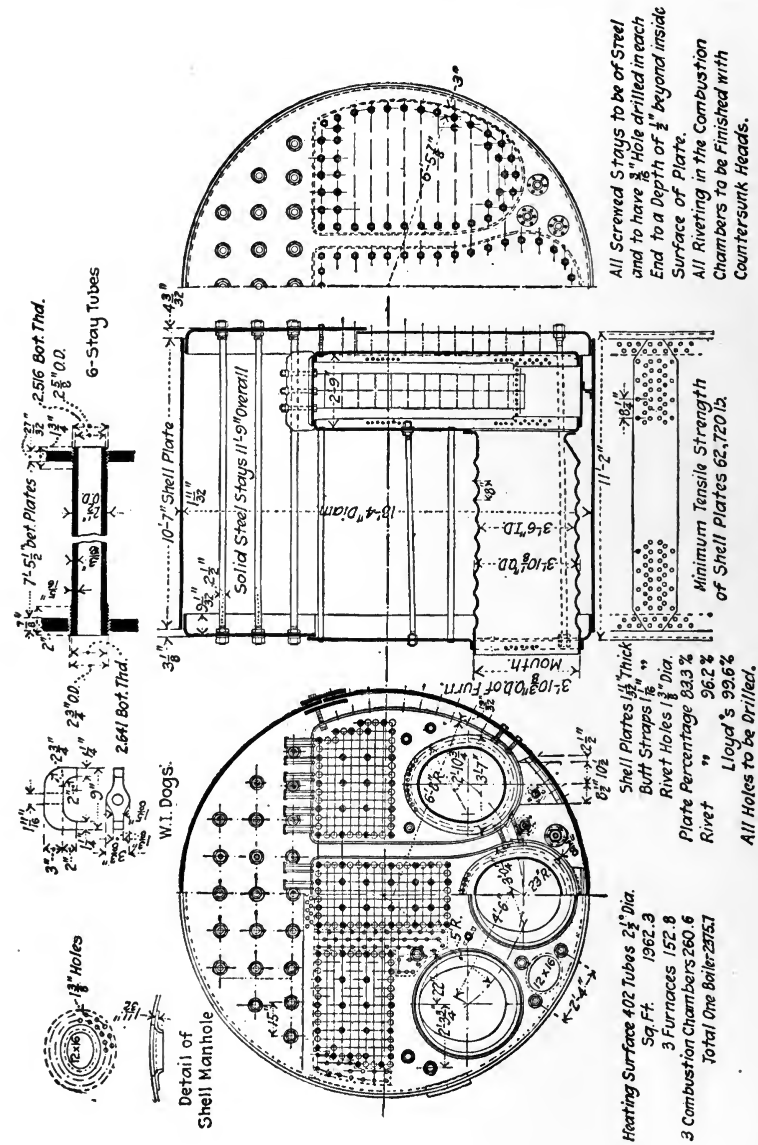


FIG. 10.—Scotch boiler with separate combustion chambers.

to the front head. As shown in the drawings and figures, the furnaces, combustion chamber, and boiler tubes are completely

surrounded by water. From the furnace, the products of combustion pass into the combustion chamber and thence through the boiler tubes to the uptake on the front of the boiler.

All the flat surfaces of the Scotch boiler, subject to pressure, which include the front and back heads, the front, back, side, and top surfaces of the combustion chamber, must be thoroughly

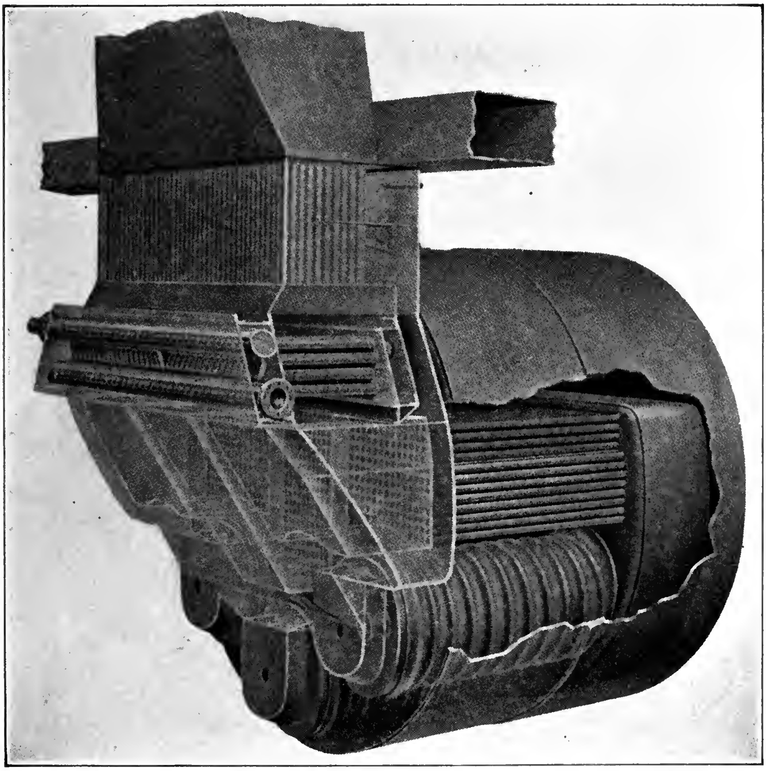

Fig. 11.—Scotch boiler with waste type superheater and Howden air heater.

stayed to prevent collapse. In the steam space solid steel stays extend from the front to the back head. These extend through the heads and are secured with nuts on both sides of the plate. The stays in the steam space must be placed far enough apart to allow for inspection and cleaning. The boiler tubes are expanded into the front head plate and the combustion chamber tube plate and are not secured well enough to stiffen these two

flat plates. To stiffen these flat plates, stay tubes, which are extra thick boiler tubes with screwed ends, are placed at regular intervals among the boiler tubes.

The back head and the back sheet of the combustion chamber are fastened together by screwed stay bolts. The top of the

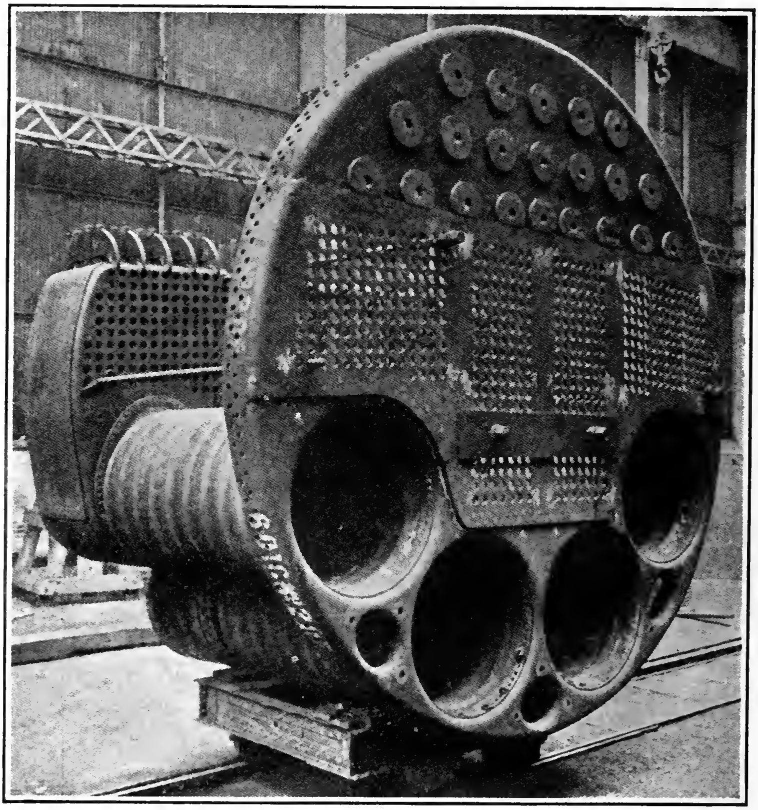

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Fig. 12.—Scotch boiler under construction—front view.

combustion chamber is supported by means of longitudinal girders (Figs. 12 and 13) which are attached to the flat plate by screw bolts. These girders bear on the front tube plate and back plate of the combustion chamber and transmit the load to these members.

The furnaces are corrugated both to strengthen them against collapse and to provide for longitudinal expansion. The Mor-

rison suspension furnace, so called because the metal between the corrugations is in the form of a catenary, is used almost universally in the United States.

The above gives a brief description of the construction of the Scotch boiler and the various features mentioned should be

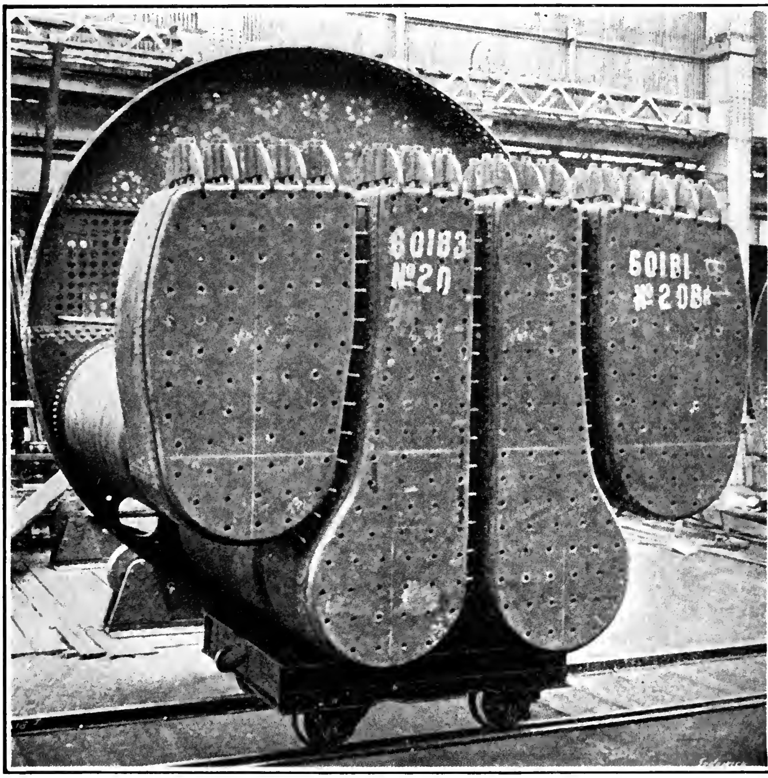


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FIG. 13.—Scotch boiler under construction—rear view.

carefully studied from the drawing, Fig. 10. Rules for the construction of Scotch boilers are published by the U. S. Board of Supervising Inspectors of Steam Vessels, the British Board of Trade, and the various classification societies such as the American Bureau of Shipping and Lloyd's Register of Shipping. All marine boilers must comply with some set of rules. A summary of these rules is given in Sterling's Marine Engineers' Handbook.

Scotch boilers may be divided into two types: single ended (Fig. 10) and double ended boilers (Frontispiece). Each of these two types may be built with separate or common combustion chambers. A single ended boiler with two or more furnaces may have a separate combustion chamber for each furnace as shown in Figs. 10, 12 and 13, or all the furnaces may be attached to one common combustion chamber. The boiler with separate chambers is the one almost universally used. While the separate combustion chamber type is more expensive than the common combustion chamber type it has the following advantages: (1)

Table IV.—Weights of Marine Boilers

| Make | H.S. | Installation | Sq. ft. h.s. per cu. ft. space | Weight of boiler and water, lbs./sq. ft. h.s. | Authority |
|---|---|---|---|---|---|
| Babcock & Wilcox | 5,359 | U.S.S. Utah | 2.29 | 23.70[1] | Marine Steam |
| Babcock & Wilcox | 2,000–4,000 | Average values | ........ | 30–40 | 4 in. tube type |
| Fore River Yarrow | 6,614 | U.S.S. Duncan | 2.58 | 11.60 | S.N.E. May, 1914 |
| Normand | 4,780 | U.S.S. Trippe | 1.79 | 12.40 | Marine Steam |
| White Forster | 4,500 | U.S.S. Mayrant | 2.66 | 12.10 | Marine Steam |
| Almy | 438–1,098 | Class C boiler | ........ | 19–24[1] | Marine Steam |
| Scotch | 2,500–3,500 | Average values | ........ | 65–70[1] | |

[1] Coal burning.

due to the separation of the tubes into banks, the boiler is much more accessible for cleaning and inspection and (2) it can be adapted to forced draft while the other type cannot be, except with difficulties in firing.

Small Scotch boilers for tugs, small coastwise ships and harbor vessels frequently have a common combustion chamber. This is generally the case, also, with boilers having only two furnaces. Double ended boilers are not used very largely except where a large number of boilers are required.[1] It has been found by experience that the double ended boiler requires more repairs and is shorter lived than the single ended boiler. Now it is more the universal practice to fit two single ended boilers back to back; although this arrangement takes up more fore and aft space. Quite frequently double ended boilers are built with a common combustion chamber for the two corresponding fur-

[1] See boiler installation on S.S. "Mauretania." *Engineering*, London, 1907.

naces in the opposite ends of the boiler. This practice cannot be adopted where forced draft is used, for the draft from one furnace would blow the fire out into the other fireroom when the door in the opposite furnace was opened.

If double ended boilers were used in ships of small boiler

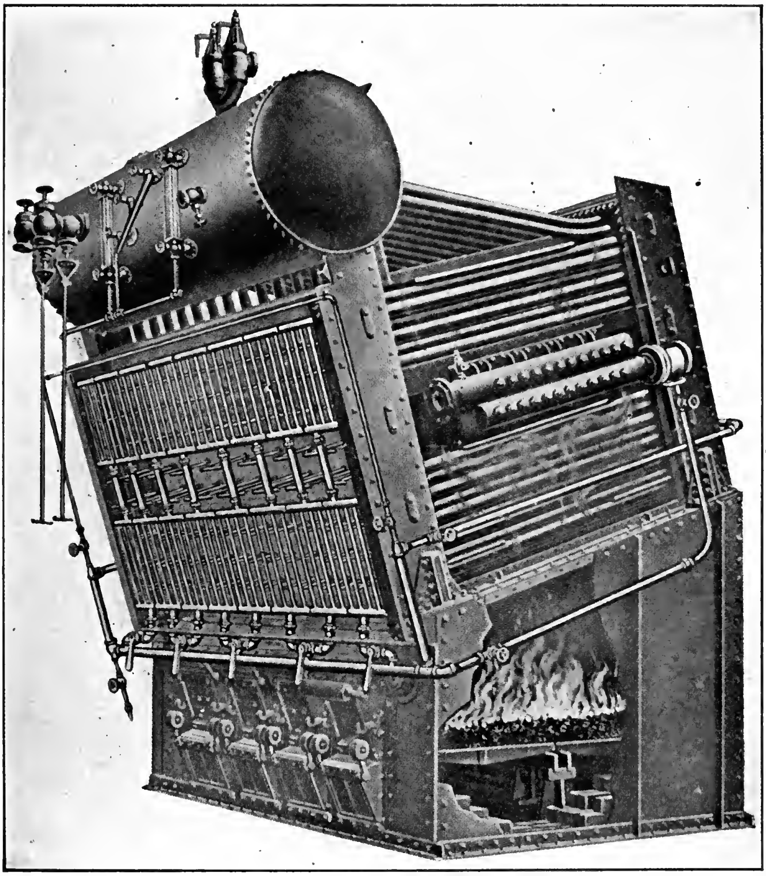


Fig. 14.—Foster boiler fitted with Diamond soot blowers.

power in place of two single ended boilers, an accident to one boiler might seriously curtail the boiler power as the equivalent of two single ended boilers would have to be cut out.

Scotch boilers run from 10 ft. in diameter and 10 ft. long with two furnaces and 1,000 sq. ft. of heating surface up to

17 ft. in diameter and 12 ft. long with four furnaces and 3,500 sq. ft. of heating surface.

A comparison of the Scotch boiler with the water-tube boiler is given in Art. 44; weights of Scotch boilers are given in Table IV; and test data are given in Tables V and VI.

**34. Water-tube Boilers.**—Water-tube boilers, as the name implies, are distinguished from Scotch or fire-tube boilers, in

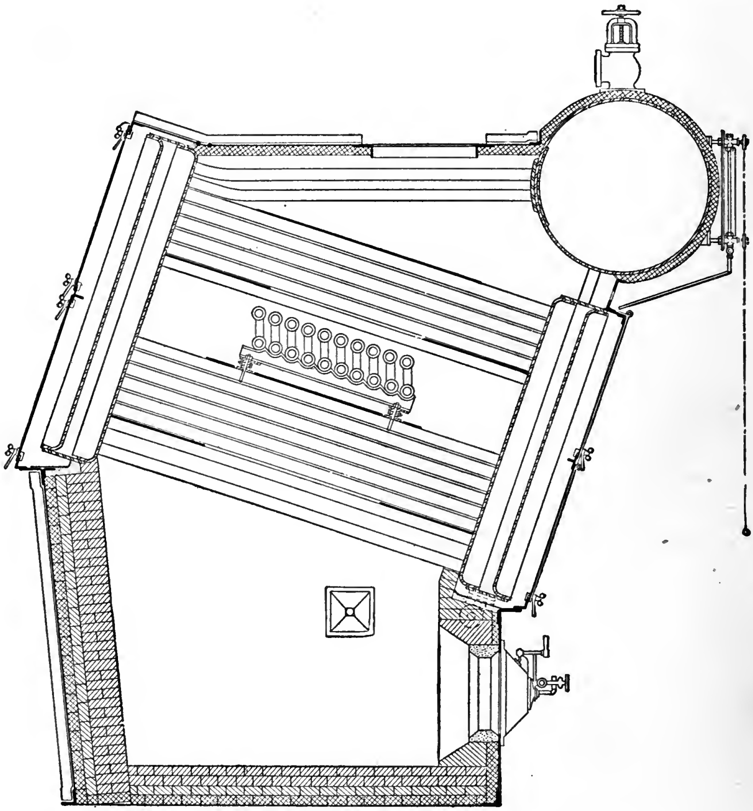


Fig. 15.—Foster water-tube boiler fitted for oil burning.

that the water circulates through the boiler tubes instead of surrounding them on the outside. Water-tube boilers are further characterized by the small volume of water that they contain and by their light weight. A description of the water-tube boiler can best be given by taking up several representative types in detail.

**35. Foster Water-tube Boiler.**—Illustrations of this boiler are given in Figs. 14 and 15. The boiler consists of a large number of closely spaced 3-in. tubes fitted into front and rear headers; the headers are in turn connected to a horizontal steam and water drum fitted on the front of the boiler. The headers are built up of steel plates securely stiffened by hollow stay bolts. Alternate tubes are left out of every second row to allow for these stay bolts. These hollow stay bolts besides stiffening the header plates afford an opening for inserting the nozzle of the soot blowers (Fig. 14), thus insuring efficient cleaning of the tubes. The headers are 8 in. deep and are connected to the cross drum as shown in Fig. 15. With the drum half filled the water capacity of drum, headers, and tubes is sufficient to allow about 15 minutes steaming without additional feed water.

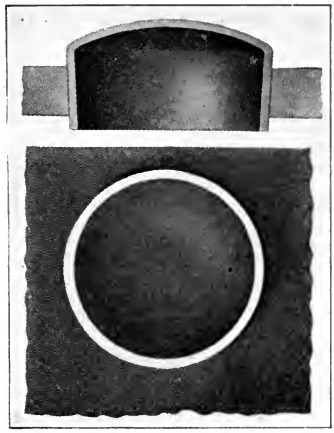


FIG. 16.—Handhole cap Foster boiler.

A special feature of the Foster boiler is the metal to metal joint used in the handhole caps, Fig. 16. These handholes are

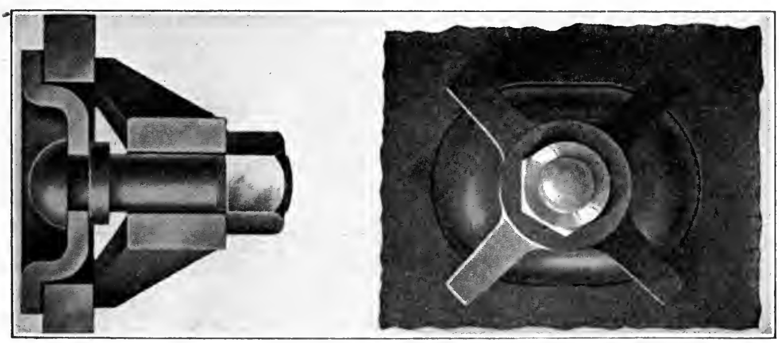


FIG. 16a.—Master handhole closure used at bottom of headers.

cut in the outer header plate opposite each tube to allow for the renewal of a tube, cleaning and inspection; and each one must be closed by a cap before filling the boiler. The handhole

openings are circular and slightly tapered. The caps are inserted into the header through master handholes at the bottom of each header and are placed in the handholes by a special pickup tool. They are then drawn into place from the outside by a special tool, and are removed by applying a pressure from the outside. This metal to metal joint eliminates all gaskets and absolutely insures that the joints cannot open up and leak under pressure.

Metal baffle plates are fitted to insure that the tubes are completely swept by the hot gases. The Foster boilers, however, rely more on the close spacing of the boiler tubes and a high gas velocity to give an efficient heating surface than on baffling. The group of cross tubes passing at right angles through the middle of the tubes in Fig. 15 is the superheater and will be discussed in a later chapter. The furnace insulation of the Foster boiler consists of two layers of 2 1/2 in. insulating brick next to the casing and a 9-in. layer of firebrick inside of the insulating brick. To secure the brickwork against vibrations and action of the ship in a seaway special bricks are fitted with hook bolts to hold the bricks to the casing (Fig. 15).

The Foster type of boiler has been fitted in a large number of merchant ships during the past few years and with proper handling is giving excellent service.

**36. The Babcock & Wilcox Water-tube Boiler.**—The Babcock & Wilcox Co. makes two main types of boilers, a 2-in. tube type used on naval vessels and high speed passenger ships and a heavier and more rugged design with 4-in. tubes used on merchant ships. The former, due to the smaller tubes, has more heating surface for given boiler dimensions than the latter and hence is used where light weight is important. The cleaning of both the inside and outside of the tubes is much easier in the boiler with the larger tubes.

The Babcock & Wilcox boiler is used practically universally in the battleships of the U. S. and foreign navies and the 4-in. tube type has been fitted to a considerable extent in merchant ships during the past few years.

The following description of the Babcock & Wilcox marine boiler is taken from the B & W publication, *Marine Steam.* The arrangement and construction of the boiler are clearly shown in Fig. 17 for coal burning and Fig. 18 for oil burning.

The tubes forming the heating surface are divided into vertical sections and, to insure a continuous circulation in one direction, are placed at an inclination of 15 degrees with the horizontal. By distributing the surface

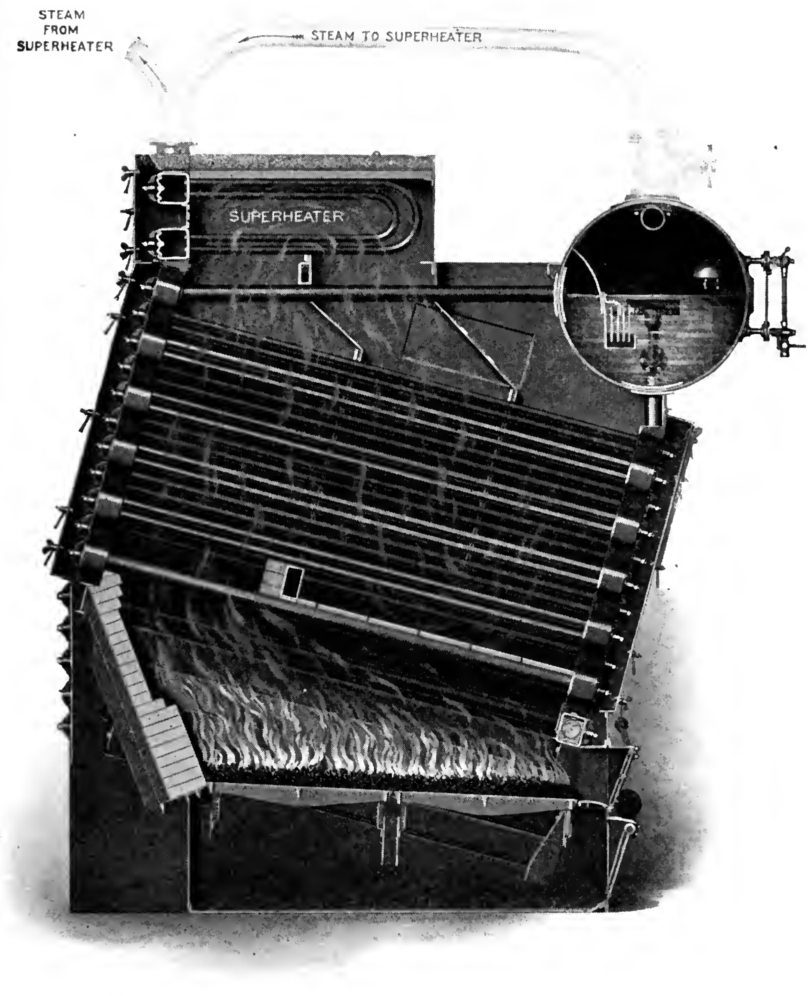



Fig. 17.—Babcock & Wilcox water-tube boiler.

into sectional elements, all danger from unequal expansion due to raising steam quickly, or to sudden cooling, is at once overcome.

Each section is made up of a series of straight tubes expanded at their

ends into corrugated wrought-steel boxes known as headers. As the headers are staggered, the tubes are so disposed that lanes for the sudden escape of the products of combustion are prevented. The hot gases are therefore completely broken up in their passage across the heating surface.

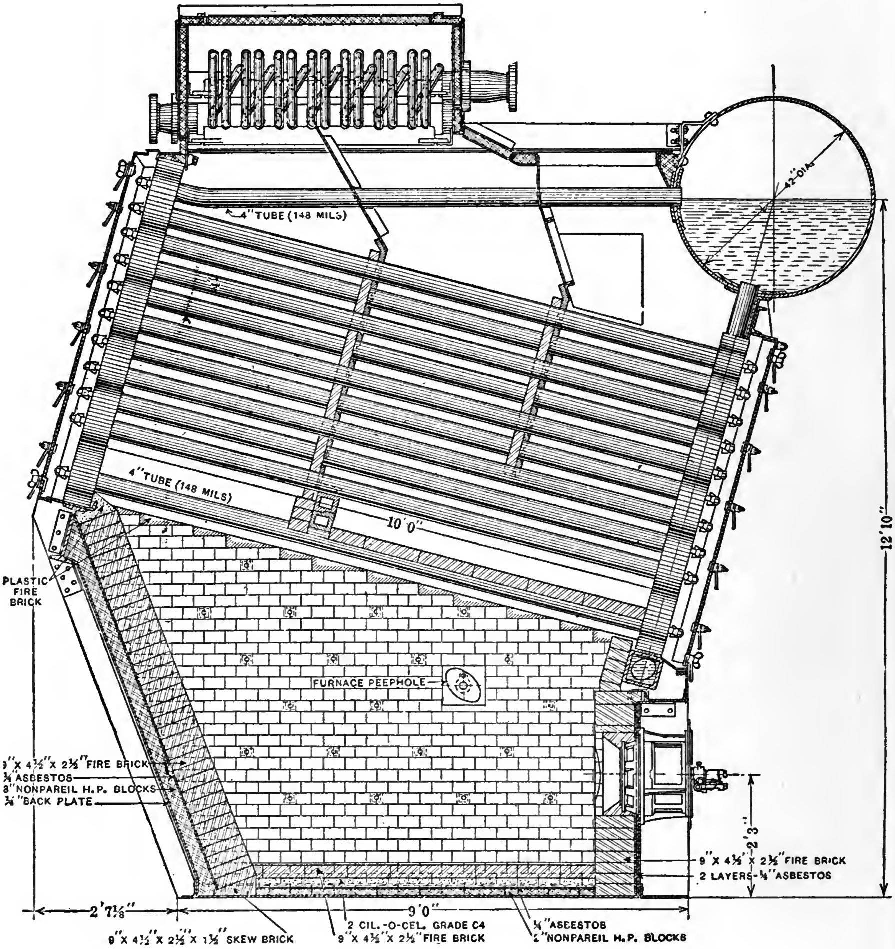



FIG. 18.—Babcock & Wilcox boiler fitted for burning oil.

The side sections are continued down to the level of the grate, the tubes being replaced by forged steel boxes of 6 in. square section at the furnace sides. These boxes are located one above the other at the same angle as the tubes; they take the place of brickwork, ensure a cool side

casing; prevent the adherence of clinkers, and are of sufficient thickness to withstand the wear and tear of the fire tools.

Extending across the front of the boiler and connected to the upper ends of the headers is a horizontal steam and water drum of ample dimensions. As the upper ends of the rear headers are also connected to this drum by horizontal tubes, each section is provided with an entirely independent inlet and outlet for water and steam.

Placed across the bottom of the front header ends and connected thereto by 4-in. tubes, is a forged steel box of 6-in. square section. This box, situated at the lowest corner of the bank of tubes, forms a blow-off connection or mud drum through which the boiler may be completely drained.

The circulation of the water is as follows: Heat being applied to the inclined tubes and vapor formed, the water and steam rise to the high end and flow through the up-take headers and horizontal return tubes to the steam and water drum, the path of both water and steam being short and direct; the water evaporated in the tubes being replaced by water flowing directly from the bottom of the drum downward through the front headers, or down-takes.

Upon entering the drum, the steam and circulating water are directed against a baffle plate, which causes the water to be thrown downward, while the steam separates and passes around the ends of the baffle plate to the steam space, from which it is conducted by a perforated dry pipe to the stop valve.

By a roof of light firetile, supported upon the lower tubes and extending part way over the furnace, the gases evolved from fresh fuel are compelled to flow toward the rear of the boiler, passing over an incandescent bed of coals and under the hot tile roof.

As the furnace increases in height approaching the bridge wall, the gases have both space and time in which to mix thoroughly and burn before entering the bank of tubes forming the heating surface. By this arrangement a high furnace temperature is established, which is acknowledged by all authorities to be the essential requirement of boiler economy.

The circuitous route which the gases are compelled to follow, in crossing the heating surface three times before exit, causes them to impart to the tubes the greatest possible amount of heat.

**37. Ward Water-tube Boiler.**—The Ward water-tube boiler, Fig. 19, is similar in general characteristics to the Babcock & Wilcox and Foster boilers. The main feature of the boiler that differentiates it from the B & W and Foster is the construction of the headers. These are built up of two flat plates as in the Foster boiler but instead of using screwed stay bolts for stiffening

the flat plates of the headers, solid diaphragm plates are used extending the whole length of the header. These diaphragm plates are secured to the heater plates by a slotted T construction as shown in Fig. 20. The header is thus divided into vertical compartments, two tubes wide. The handholes are made large enough to take care of the removal of four tubes. The baffling

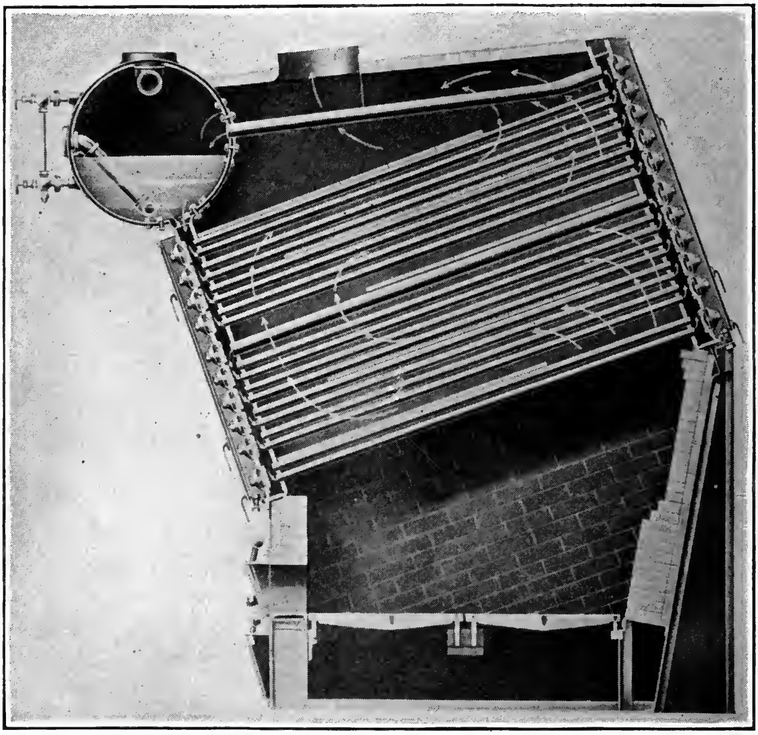


Fig. 19.—Ward water-tube boiler.

is somewhat special, the path of the gases being shown in Fig. 19. A number of Ward boilers have been built with 60-in. and 54-in. steam drums instead of a 42-in. drum. The boilers with the drum of larger diameter are slightly heavier than those with smaller drums due to the larger volume of water. These boilers, however, have found considerable favor among the operating personnel because the larger volume of water carried requires less attention on the part of the fireroom force. The

slight increase in weight of the large drum boiler is of no serious concern when they are fitted on cargo ships.

**38. Yarrow Water-tube Boilers.**—This boiler is representative of the light weight "express" type of water-tube boilers. It is used almost exclusively on high speed naval vessels, although it has been used to some extent on passenger vessels. This type of boiler has been fitted on the "Vaterland" and "Imperator"[1] and also on the large passenger vessels built for the Emergency Fleet Corp. Table IV, which gives weights of various types of

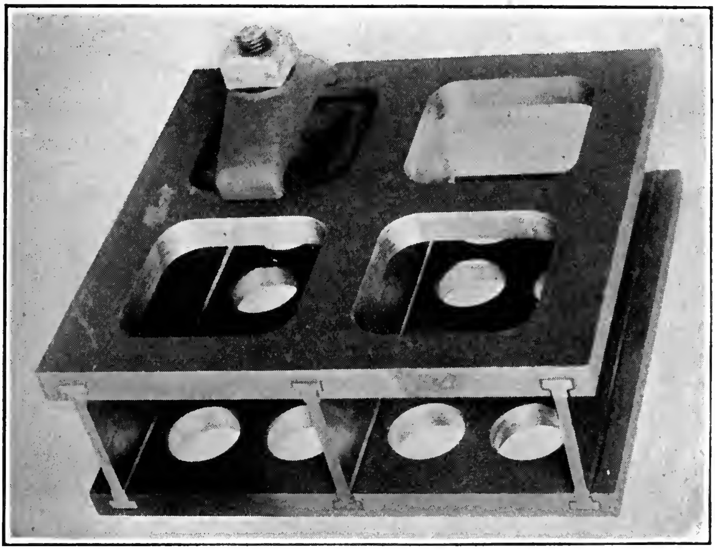


FIG. 20.—Header construction of Ward boiler.

boilers, shows the light weight of these boilers compared with the Scotch and other types of water-tube boilers.

The Yarrow boiler as shown in Figs. 21 and 22 consists of three drums, two small ones at the bottom connected by tubes to a larger one at the top. The furnace is located at the center between the tubes directly beneath the upper drum. In the latest design of the Yarrow boiler the feed enters the upper drum as shown. In the earlier types the feed water was introduced into the lower drums. The lower drums are entirely filled with water; the water level is at about the center of the upper drum.

[1] Now "Leviathan" and "Berengaria."

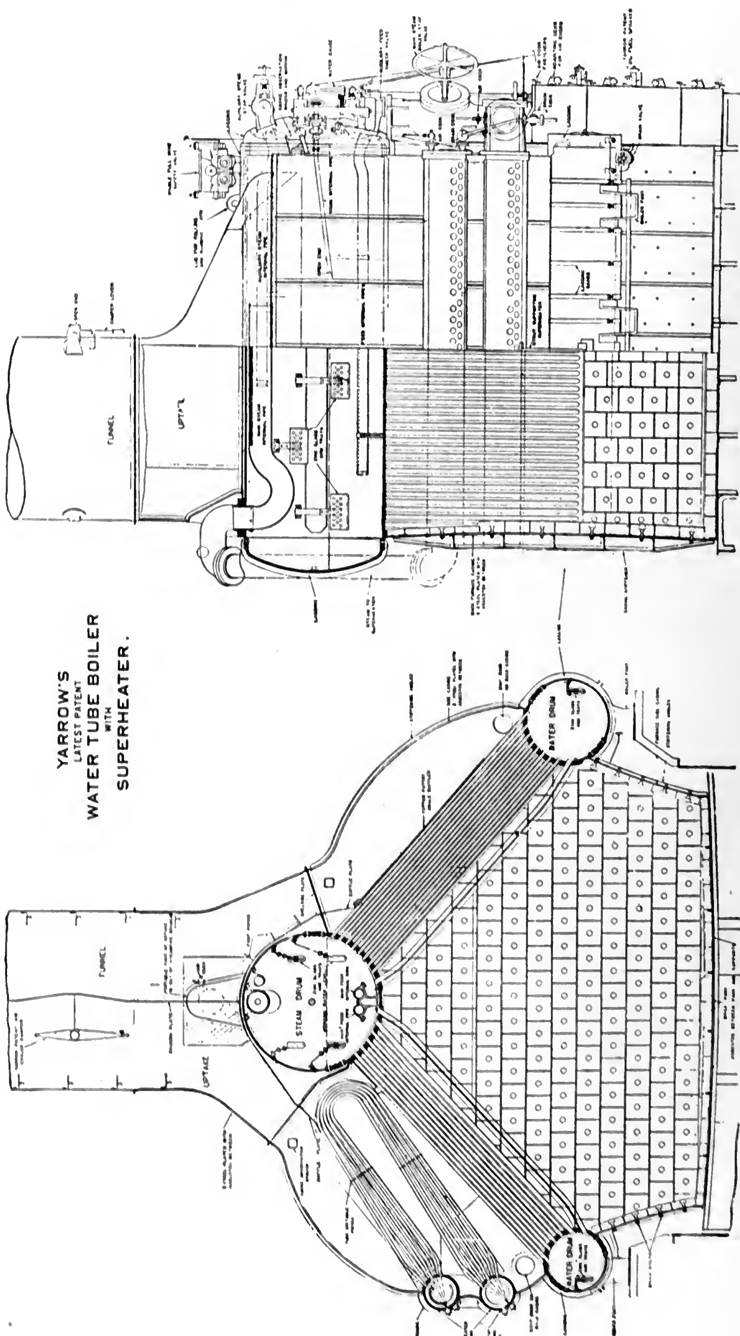



FIG. 21.—Yarrow water-tube boiler.

Some of the tubes serve to carry the hotter water up and others to carry the cooler water down. No definite tubes are used for either purpose, the circulation being different at different times. The hot gases make but one pass through the tubes and then escape up the uptake. Baffles are fitted, as shown in the illus-

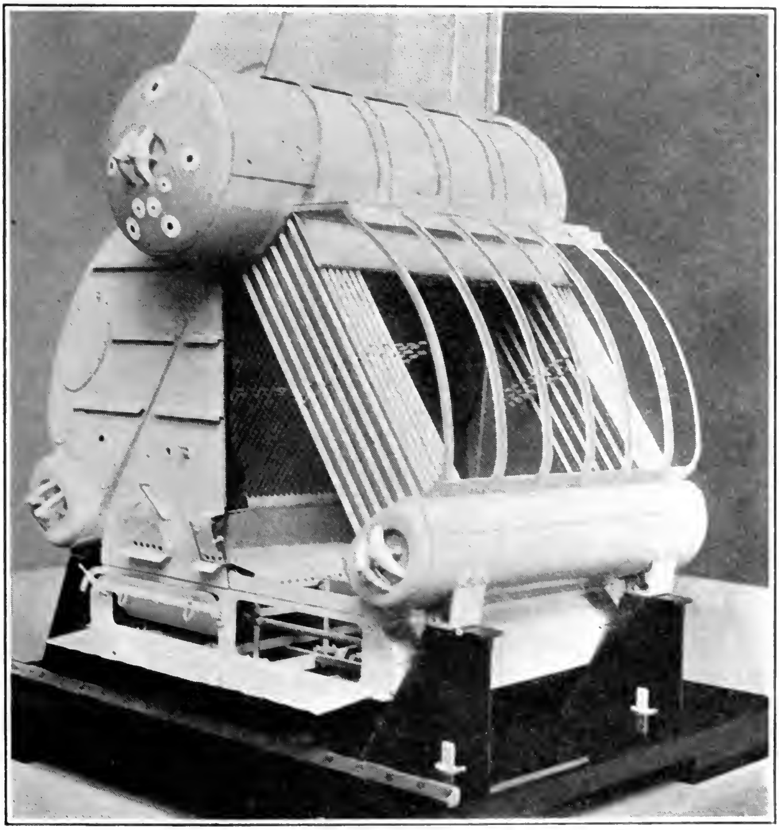


FIG. 22.—Model of Yarrow boiler.

tration, to direct the gases through the tubes. The tubes which are 1 in. and 1 1/4 in. diameter are small compared to those used in other types of water-tube boilers and are closely spaced. Defective tubes can only be replaced by cutting out the adjacent ones. Further details of this boiler when fitted with a superheater are given in Art. 81.

**39. Heating Surface and Heat Transmission.**—The heating surface of a boiler consists of all the surface which has hot gases on one side and water on the other, and transmits the heat from the hot gases to the water. The heating surface of a Scotch boiler is made up of the furnaces, the combustion chamber and the tubes; that of a water-tube boiler is made up of the outside surface of the tubes and those parts of the drum and headers exposed to the hot gases.

The rate at which heat is transmitted from the gases to the water depends on: (1) the conductivity of the material of which the heating surface is composed, (2) the difference in temperature

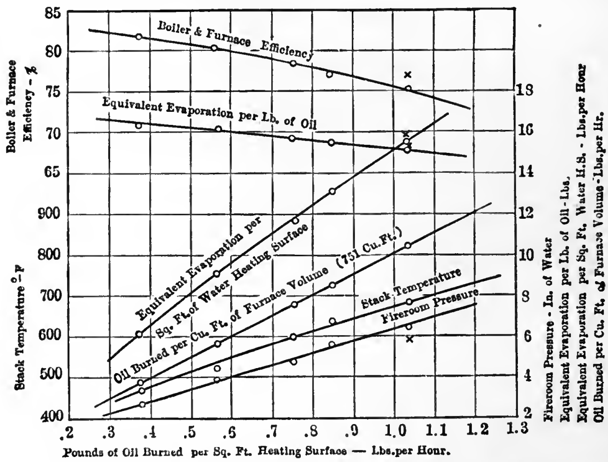


Fig. 23.—Performance curves of White-Forster boiler.

between the gases and the water; (3) the velocity of the hot gases over the heating surface; (4) the velocity of the water over the heating surface; and (5) the cleanliness of the heating surface which depends on the amount of oil or scale on the water side and the amount of soot on the side exposed to the gases. The *rate of transmission* is expressed in b.t.u.'s per sq. ft. per degree difference per hour.

It is obvious, due to the large difference in temperature between the gases and water, that the portion of the heating surface adjacent to the furnace is transmitting heat at a very rapid rate while other parts of the heating surface such as the tubes in the last pass of the gases are transmitting heat at a very low rate. It is possible to increase the area of the heating surface

to such an extent that the temperature of the gases will be reduced to the temperature of the steam and water in the boiler. The size, weight, and cost of a boiler,—all important characteristics for a marine installation,—depend directly upon the amount of heating surface. A high efficiency requires a large heating surface in order to reduce the temperature of the gases leaving the boiler. Thus we see that the size, cost and weight of a boiler and the economical performance work in opposition to one another. It is the duty of the boiler designer to proportion his heating surface so that the best all around boiler will result, giving a proper balance between size and

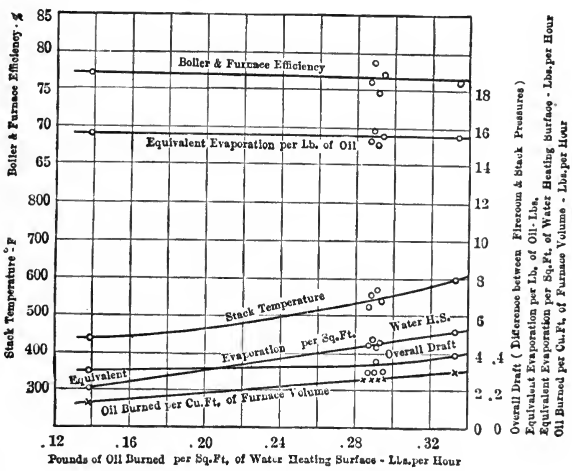



FIG. 24.—Performance of B. & W. boiler with natural draft.

economy. This is accomplished by designing the boiler so that the gases leave the boiler under service condition at 100 to 150° above the temperature of the steam. For a boiler under 250 lb. gage pressure (405°F.) this would give a temperature at the base of the stack between 500 and 550°F. For high speed vessels where weight is of extreme importance, the economy is sacrificed somewhat by allowing a greater final temperature difference between the steam and the escaping gases under full power conditions. Thus, while the temperature at the base of the stack for water-tube boilers in merchant ships is between 500 and 600°, high speed naval boilers operating at maximum boiler capacity may have temperatures at the base

of the stack as high as 700 and 800°. The steam temperature for both cases is in the neighborhood of 400°F. The reader should consult Figs. 23 and 24 which show the performance curves of two types of boilers.

The temperature of the gases leaving a Scotch boiler are much higher than for water-tube boilers under similar conditions. This is due to the short length used with Scotch boilers. The temperature of the stack gases can be reduced and the efficiency of the boiler increased by adding more heating surface at the place where the gases leave the boiler. This means increasing the length of the boiler tubes and consequently the length of the boiler. Retarders are frequently fitted in the tubes of Scotch boilers to increase the rate of heat transmission by

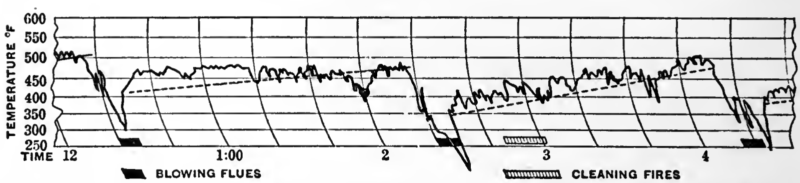



Fig. 25.—Reduction in stack temperatures by blowing soot from tubes—Foster boiler.

agitating the hot gases and bringing them in better contact with the tube heating surface. Retarders consist of flat pieces of steel twisted into the form of screw threads of large pitch. These retarders are about ¼ in. smaller than the diameter of the tubes and extend the whole length of the tube. Retarders are used frequently with coal fired boilers and almost universally with oil fired boilers. The amount of agitation of the gases is controlled by the number of turns given to the steel bar which forms the retarder.

The common practice with Scotch boilers is to use a short boiler and fit Howden air heaters, waste type superheaters, or both, in the uptake. These uptake auxiliaries absorb the heat in the waste gases and reduce the temperature at the base of the stack to 500° or even lower. Without these additional auxiliaries the efficiency of the Scotch boiler in service conditions would be considerably below that of the water-tube boiler. The overall economy of the boiler when fitted with the Howden heater for heating the air used for combustion or the waste type superheater is practically the same as that of

the water-tube boiler.[1] Fig. 32 shows a Scotch boiler fitted with the Howden air heater and Fig. 11 with a Howden heater and a waste type superheater. Fig. 27a shows the results of temperature measurements in a Scotch boiler fitted with air heater and superheater in the uptake and should be consulted at this point. The superheater and Howden heater will be considered in detail in a later chapter.

It has often been proposed and to some extent tried, to construct a boiler on the counter flow principle so that the cold feed water will come in contact with the coolest gases in the upper part of the boiler and then flow downward so that the heated water will be in contact with the hotter gases lower in the boiler. This would allow a larger temperature difference between the gases and water at the upper part of the boiler and greatly increase the effectiveness of the heating surface. For example, if the gases were at 500° and the feed at 200° the temperature difference in the upper part of the boiler would be 300° instead of 100° in the case already mentioned where the steam was at 250 lbs. pressure.

This same principle is used in land installations by fitting feed water heaters, known as "economizers," in the flues between the boiler and the stack. Because of the added resistance caused by the economizer, forced draft always has to be used when they are installed. Tests are recorded of boilers fitted with economizers and forced draft where the temperature of the gases leaving the boiler was reduced from 490 to 240° at the exit of the economizer. Economizers have never been used on shipboard because of their excessive weight and the space occupied. This is clearly brought out by an example in Art. **182**.

As already pointed out, the rate of transmission of heat to the water, and consequently the effectiveness of the heating surface, depends upon the velocity of the gases, the circulation of the water and the cleanliness of the heating surface. High gas velocity not only brings a larger amount of heat in contact with the tubes in a given time but it also agitates the gases

[1] Tests were carried out in 1921 by the U. S. Shipping Board on a Scotch boiler with fuel oil. The boiler was fitted with retarders, Howden air heater, and waste type superheater. The efficiencies of the boiler and superheater were between 80 and 84.7 per cent with retarders, and considerably lower without retarders.

surrounding the heating surface and causes a better contact between the heating surface and gases. This is accomplished by the use of closely spaced and staggered tubes and baffles in water-tube boilers and retarders in Scotch boilers.

All types of water-tube boilers have excellent water circulation due to the small amount of water present, and the fundamental features of design by which the water is broken up into small paths. Fresh supplies of water are constantly brought in contact with the hottest part of the heating surface. The rapid circulation of water in water-tube boilers is the main reason why steam can be raised quickly, and high rates of evaporation maintained.

The necessity of keeping the heating surface clean on both sides is very important. Soot will accumulate rapidly on the tubes especially with coal fired boilers and must be frequently removed either by hand methods or by mechanical soot blowers if high efficiency is to be secured. Figure 25 taken from the catalog of the Foster Marine Boiler Corp. shows the effect of the accumulation of soot in service on the temperature of the escaping gases. Carbon in the form of soot is one of the best non-conductors known. A 1/16-in. thickness of soot on 1/8-in. plate will decrease the heat conductivity about 25 per cent. Tests of boilers with and without mechanical soot blowers show an increase in efficiency of 2 to 4 per cent for the boilers fitted with the soot blowers. A gradual rise in the temperature of the stack gases is an indication of the accumulation of soot on the tubes. The necessity of keeping the surface in contact with the water free from scale, sediment, oil and grease, requires careful attention to the chemical composition and quality of the feed water used. To obviate the deposit of scale on the heating surface due to the foreign matter in solution or in suspension in the water, the use of evaporators and distillers for making up feed from sea water is becoming common practice. The elimination as far as possible of all internal lubrication in reciprocating engines and the careful filtering of the feed water are necessary to keep oil out of the boilers. Oil and scale make a very non-conductive film on the inside of the heating surface and this decreases the efficiency of the boiler. Scale also reduces the capacity of the heating surface to conduct away the heat applied to it, resulting in overheating of the metal which often causes serious accidents to boilers.

The weight of a boiler is always expressed in pounds per square foot of heating surface. Table IV gives the weights per sq. ft. of heating surface of various types of boilers. The weight of a Scotch boiler naturally varies with the pressure carried; weights range between 65 and 75 lbs. per sq. ft. of heating surface "wet," i.e., including water. The B & W naval type (2-in. tubes) weighs between 20 and 25 lbs. per sq. ft. of heating surface including water, while the heavy merchant type (4-in. tubes) weighs between 30 and 45 lbs. per sq. ft. of heating surface including water. The light "express" type ranges between 10 and 13 lbs. per sq. ft. including water.

The weight of water carried in the various boilers is as follows:

Scotch, 20–25 lbs. per sq. ft. of heating surface.
Large tube, water-tube boilers (B & W) type, 5 to 7 lbs. per sq. ft.
Express type, 1.5 to 3.0 lbs. per sq. ft.

The above figures showing the water contained in the boiler per sq. ft. of heating surface is a very good index of the attention that a boiler requires. Water tenders and other members of a ship's engine force often condemn the water-tube boiler solely because careful watch has to be kept on the gage glasses while with a Scotch boiler this constant attention is not required because of the large volume of water carried.

**40. Boiler Performance.**—The *rate of evaporation* of a boiler is expressed in pounds of water per square foot of heating surface per hour. The average rate of evaporation for a Scotch boiler is around 5 lbs. per sq. ft. of H.S. per hour, while the rate of evaporation for water tube boilers may run as high as 18 lbs. per sq. ft. of heating surface per hour. The rate of evaporation is the average for the whole heating surface and is found by dividing the water evaporated per hour by the area of the heating surface.

In order that the performance of boilers which are operating under different pressures and feed water temperatures may be compared with one another, it is customary to reduce all performances to a common basis. The common basis chosen is the *equivalent evaporation* from feed water at 212° into steam at atmospheric pressure. This is spoken of as "the equivalent evaporation from and at 212°F."—meaning that the evaporation takes place "from" feed water at 212° into steam "at" 212°F., or at atmospheric pressure. To reduce an actual evaporation

to an equivalent evaporation all that it is necessary to do is divide the amount of heat converted into steam under actual conditions by the heat necessary to evaporate water from and at 212°, i.e., the heat of vaporization ($r$) at 212° (14.7 lbs. absolute pressure). An example will make this clear. Suppose that a boiler test showed that 9.7 lbs. of water were evaporated per sq. ft. H.S. per hour with the boiler pressure at 200 lbs. gage and the feed water at 170°F. The actual evaporation = 9.7 lbs. per sq. ft. H.S. The b.t.u. absorbed by the water per sq. ft. of H.S. per hour $= 9.7\ (r_1 + q_1 - q_2) = 9.7\ (837.9 + 361.4 - 137.9) = 10{,}295$ b.t.u. The b.t.u. absorbed if the evaporation had been from water at 212° into steam at atmospheric pressure would be $r$ at 212° or 970.4 b.t.u. per lb. Hence the equivalent evaporation from and at 212°F.

$$= \frac{9.7\ (r_1+q_1-q_2)}{r_{212}} = \frac{10{,}295}{970.4}$$

= 10.6 lbs. per sq. ft. H.S. per hour

where $r_1$ = heat of evaporation at 215 lbs. absolute

$q_1$ = heat of liquid at 215 lbs. absolute

$q_2$ = heat of liquid at 170°F.

The general procedure is to calculate what is known as the *factor of evaporation* and to multiply the actual evaporation by the factor of evaporation to obtain the equivalent evaporation. The factor of evaporation is the ratio of the heat required to evaporate 1 lb. of water into steam under actual conditions to that required from and at 212°.

For dry or wet steam the factor of evaporation =

$$\text{F.E.} = \frac{xr_1+q_1-q_2}{970.4}.$$

For superheated steam, this becomes

$$\text{F.E.} = \frac{Ct_s+r_1+q_1-q_2}{970.4}$$

Boiler performance from the economic viewpoint is expressed as the number of pounds of *water evaporated per pound of fuel.* In order to bring this value to a common basis it is also reduced to the equivalent evaporation from and at 212° per pound of coal or oil.

The rate of evaporation of a given boiler will naturally depend on the amount of fuel burned per hour. This is termed the *rate of combustion* and is expressed for coal burning boilers in pounds of *coal per square foot of grate surface per hour.* The rate of combustion is frequently based on "dry coal" instead of "coal as fired." In this case it is spoken of as dry coal per square foot of grate surface per hour. As oil burning boilers have no grate surface the rate of combustion is expressed in pounds of *oil burned per square foot of heating surface per hour.* For designing data, the rate of combustion is also expressed in pounds of oil per cubic foot of furnace volume per hour. The curves in Figs. 23 and 24 which are plotted against rate of combustion show how the rate of evaporation, and evaporation per pound of oil vary with the rate of combustion. The rate of combustion is increased in a coal burning boiler by increasing the thickness of the fire and increasing the draft to overcome the resistance offered by the increased thickness of the fire. With oil fired boilers the rate of combustion is increased by increasing the number of burners in operation or by increasing the oil pressure at the burners.

The rate of evaporation of a boiler is only limited by the rate of combustion possible. With hand fired coal burning boilers the rate of combustion is limited to the capacity of the fireroom force to handle coal. For continuous service, as in a merchant ship, this should never exceed between 20 and 22 lbs. per sq. ft. G.S. per hour. In naval vessels working under forced draft with closed stokeholes, rates of combustion up to 50 lbs. per sq. ft. G.S. per hour have been maintained for short high speed runs. For fast passenger vessels, rates of combustion as high as 26 to 28 lbs. per sq. ft. G.S. per hour are possible for runs not exceeding 10 hours. The rates of combustion used on shipboard are obviously higher than those used on shore because of the space and weight limitations. The curves in Figs. 23 and 24 should be studied here. When mechanical stokers are used higher rates of combustion and better efficiencies can be obtained than with hand firing.

With oil fired boilers the maximum rate of combustion is only limited by the number and capacity of the burners that it is possible to fit under a boiler. The present practice with oil fired boilers in the merchant service is between 0.4 and 0.6 lb.

of oil per sq. ft. H.S. per hour. Naval practice is higher than this —.70 to .80 lb. oil per sq. ft. H.S. per hour being the general practice with B & W boilers while a rate of 1.0 lb. is frequently used with express type boilers. E. H. Peabody, in a paper before the Society of Naval Architects and Marine Engineers in 1920, gave the following results of a test on a White Forster express type boiler of 7,565 sq. ft. of H.S. and 753 sq. ft. of superheating surface:

It will be noted that in the highest capacity test of June 10, 1919, there were eleven burners in operation, each atomizing 1,032 pounds of Navy Standard Oil (25.5 Baumé) per hour, giving a consumption of 1.5 pounds of oil per square foot of heating surface per hour. The evaporation of water per pound of oil from and at 212° F. was 15.14 pounds, giving an evaporation of water per square foot of heating surface per hour of 22.73 pounds from and at 212° with an efficiency of 76.15 per cent. The air pressure in the closed fireroom was 9.5 inches and the rate of combustion per cubic foot of furnace volume reached the very high figure of 15.12 pounds of oil per hour. It is believed that this test stands as a world's record for efficiency at high boiler and furnace capacity.

The above tests were made with the Babcock & Wilcox air registers and mechanical atomizers (Fig. 33). While the rate of combustion quoted above is exceedingly high, it has since been surpassed by a rate of 1.74 lbs. of oil per sq. ft. of heating surface with a small tube Babcock & Wilcox boiler.

The present practice of limiting the rate of combustion to .50 lb. per square foot of heating surface per hour is due primarily to feed water conditions. If an absolutely pure supply of feed water is available, higher rates of combustion can be used without difficulty. This will reduce the size and cost of the boiler.

With the more general adoption of evaporators for supplying the feed water a better quality of feed water may be available. Evaporators, however, should be of ample size to supply sufficient feed without forcing and should be of the double effect type.

In the same paper Mr. Peabody forecasts much higher rates of combustion for oil burning water-tube boilers on merchant ships. The largest burner installed on a merchant ship today has a capacity of about 600 lbs. of oil per hour while 1,500 and 1,800 lbs. per hour have been burned successfully in the Navy's Fuel Oil Testing Plant. For an hour's run after several hours steaming

in one of the tests at the Testing Plant the rate of burning was run up to 2,238 lbs. of oil per burner per hour.

**41. Boiler Efficiency.**—The losses in burning fuel under a boiler will be fully taken up in Chap. V after a study has been made of combustion. Here we are interested only in the definition of boiler efficiency.

Efficiency of boiler and furnace

$$= \frac{\text{Heat absorbed by boiler per lb. of fuel as fired}}{\text{b.t.u. value of one lb. of coal as fired.}}$$

The efficiency of the boiler is often based on "dry coal" instead of "coal as fired" and sometimes it is calculated on the basis of "combustible burned." In the former case weight of the moisture is eliminated from the coal and in the latter case the weight of the ash and the coal lost through the grate, together with the weight of the moisture, is eliminated. Care must be exercised in studying and comparing data of boiler tests to determine which efficiency is used and to make sure that all efficiencies compared are on a common basis.

There are other boiler efficiencies used in practice, but as these are seldom met and are not defined the same by all engineers they will not be included in this book. The efficiency based on "coal as fired" is the one commonly used. This, of course, is an overall efficiency of boiler, grate and firing and is the one of most commercial interest. For boilers using fuel oil the efficiency becomes:

$$\text{Boiler Efficiency} = \frac{\text{Heat absorbed by boiler per pound of oil burned}}{\text{b.t.u. value of one lb. of oil}}$$

The analysis of test data from an oil burning water-tube boiler will make clear the foregoing definitions.

| | |
|---|---|
| Boiler pressure (gage) | =190.3 lbs. |
| Quality | =59.2°F. superheat. |
| Feed water temperature | =64°F. |
| Oil burned per hour | = 1,300 lbs. |
| Water evaporated per hour | =15,973 lbs. |
| Boiler heating surface | = 3,050 sq. ft. |
| B.t.u. value of fuel oil | =19,311 b.t.u. per lb. |

B.t.u. absorbed by one lb. of water $= r_1 + q_1 - q_2 + ct_s. = 841.4 + 357.1 - 32.1 + (.61 \times 59.2) = 1{,}202.5$ b.t.u.

$$\text{Factor of evaporation} = \frac{1{,}202.5}{970.4} = 1.24$$

$$\text{Actual evaporation per lb. of oil as fired} = \frac{15{,}973}{1{,}300} = 12.29 \text{ lbs.}$$

Equivalent evaporation from and at 212° per lb. of oil $= 12.29 \times 1.24 = 15.24$ lbs.

Actual evaporation per sq. ft. heating surface per hour $= \frac{15,973}{3,050} = 5.24$ lbs.

Equivalent evaporation from and at 212° per sq. ft. H.S. per hour $= 5.24 \times 1.24 = 6.49$ lbs.

Boiler efficiency $= \frac{1,202.5 \times 12.29}{19,311} = 76.6$ per cent.

Rate of combustion $= \frac{1,300}{3,050} = 0.426$ lb. oil per sq. ft. heating surface per hour.

**42. Ratio of Heating Surface to Grate Surface $\frac{\text{H.S.}}{\text{G.S.}}$.**—This ratio is a value commonly tabulated for all coal burning boilers. It varies widely, not only for boilers of different types but among boilers of one make.

TABLE VII.—H.S. ÷ G.S.

| | |
|---|---|
| Scotch boilers | 30–50 |
| B & W marine type | 25–40 |
| Foster marine type | 45± |
| Almy water-tube | 35–40 |
| B & W stationary type | 55± |

Obviously the smaller the ratio of $\frac{\text{H.S.}}{\text{G.S.}}$ the smaller and more compact is the boiler. However, as the ratio increases, the weight of the boiler per square foot of heating surface becomes smaller. This is due to the fact that the heating surface increases faster than the weight for a boiler of given grate surface. This ratio is a very necessary guide in selecting a boiler as will be brought out in Chap. XIX.

**43. Boiler Horsepower.**—A unit of measurement known as a boiler horsepower is used universally in land practice. The boiler horsepower is equivalent to the evaporation of 34.5 lbs. of water from and at 212°F. (33,479 b.t.u.). This horsepower has absolutely no significance and fortunately has never been used in marine work. The size of a boiler is determined solely by the requirements of the main engines and auxiliaries. A reciprocating engine of poor economy naturally will require more boiler power than a geared turbine installation. After the steam consumption of the propelling machinery has been determined the required heating surface of the boiler can be readily fixed

upon. Calculations for the size of boilers are given in Chap. XIX. Boilers should never be spoken of or thought of in terms of the horsepower of the main engine but in terms of heating surface.

**44. Comparison of Scotch and Water-Tube Boilers.**—Both the Scotch and water-tube boilers have merits and defects. The Scotch boiler has been almost universally used in the merchant service; all sea going engineers are familiar with it; firemen can easily be secured capable of operating it; repairs can be made in practically every port of the world; and it has a large water volume and consequently is not as dangerous in the hands of an inexperienced crew as a water-tube boiler.

The claim is frequently brought forward for Scotch boilers that they can be used successfully when there is salt water in the feed system. This is true provided sufficient volume is given the steam space to prevent priming. This argument is sometimes carried further, by declaring Scotch boilers are often necessary as salt water cannot be kept out of the feed system. With the present designs of condensers and proper attention to tube packing there is no excuse whatsoever for salt water to leak into the condensate; so this argument should be given little weight.

The above comprise the merits of the Scotch boiler. Its defects and disadvantages are many. It has a very poor circulation and steam must be raised slowly. As a result of the poor circulation unequal stresses are set up often causing it to be in a leaky condition and necessitating frequent and expensive repairs. On account of the long time required to raise steam it is particularly bad on short runs where it is frequently cooled down and heated up. Very often the time required for cooling, cleaning or repairing, and raising steam again in a Scotch boiler, decides the length of stay in port. Scotch boilers cannot be forced as a water-tube boiler can and are apt to be short lived if an evaporation rate in service of much over 5 lbs. / sq. ft. of heating surface is maintained.

Because of the severe stresses set up necessitating heavy construction, the Scotch boiler is practically limited to steam pressures below 200 lbs. while water-tube boilers can be built to carry more than twice this pressure. For the same reasons, the size of a single ended Scotch boiler is limited to about 4,000

sq. ft. of heating surface while water-tube boilers of 10,000 sq. ft. are now under construction.

Unless the Scotch boiler is fitted with a waste type superheater and Howden air heater in the uptakes, or fitted with retarders and given a long length of tube to reduce the temperature of the escaping gases, its efficiency will be less than that of the water-tube boiler. This is especially true if the boiler is operated on fuel oil at high rates of combustion.

The water-tube boiler has many advantages over the Scotch boiler. It occupies less space; costs much less for a boiler of equal capacity; the repairs and upkeep charges are lower; and it is much lighter. Further, the water-tube boiler has an excellent circulation and small water volume and can be easily forced without any serious results; steam can be raised quickly and it can be quickly cooled for repairs. Thus a case will seldom arise where a ship's stay in port is influenced by the boilers. As higher steam pressure can be carried with a water-tube boiler, this alone may often be enough to cause the shipowner to decide in favor of this type.

A case is on record where a water-tube boiler was blown down, a tube removed and steam raised again in 2 hours. While it is possible to raise steam in a light water-tube boiler in 15 minutes, it is best to allow several hours for heating up the brickwork and starting the circulation with a cold boiler when sufficient time is available. With a Scotch boiler 24 hours should be allowed for raising steam if possible. Steam should never be raised in less than 5 or 6 hours and an equal amount of time should be allowed in cooling down. To attempt to raise steam or cool a Scotch boiler in too short time will bring undue strain on the boiler because of the poor circulation, and a leaky boiler will result.

From the foregoing statements, it would seem that the tradition of the merchant service, its general familiarity, and ease of handling with an inexperienced crew are the only things which recommend the Scotch boiler. For the most efficient operation and reduced operating costs of both ship and machinery the water-tube boiler should be more universally adopted.

The universal adoption and good service rendered by water-tube boilers in the U. S. and other navies should be sufficient recommendation for its use in the merchant service. On account

of the great saving and weight, it should be the only type considered for high powered passenger ships.

J. J. Nelis, formerly senior boiler engineer of the Emergency Fleet Corp., has given the following statement in "Marine Engineering" for January, 1920: "Practically 80 per cent of the boiler repairs on vessels of the emergency fleet that are now in operation are being made on Scotch boilers, although only 50 per cent of the boilers in operation are of this type. In many cases leaks due to poor workmanship of Scotch boilers cannot be made tight by caulking and various kinds of welding have been resorted to. Welding even when properly done does not keep seams of Scotch boilers tight for more than 4 or 5 years, due to the constant movement of these seams." It should be borne in mind, however, in connection with the above statement that many of the boilers built for the Emergency Fleet Corp. were built under war conditions by inexperienced men and not by shipbuilders with long experience with Scotch boilers.

**Table V.—Performance of Coal Fired Boilers**

| Make | Scotch D.E. | Scotch S.E. | Foster W.T. | | | | E.F. Corp. W.T. | | 2-in. Tube Babcock & Wilcox | |
|---|---|---|---|---|---|---|---|---|---|---|
| Reference | Sterling[1] | Sterling[2] | Foster catalog | | | | A.S.M.E. 1919 | | B. & W. catalog | |
| Grate surface | 139 | 59 | 65.8 | 65.8 | 65.8 | 65.8 | 66.5 | 77.5 | 57.9 | 57.9 |
| Heating surface | 3,455 | 2,340 | 3,050 | 3,050 | 3,050 | 3,050 | 2,500 | 2,500 | 2,571 | 2,571 |
| Superheating surface | ........ | ........ | 338 | 338 | 338 | 338 | 0 | 0 | 0 | 0 |
| Ratio h.s./g.s. | 24.8 | 39.75 | 46.3 | 46.3 | 46.3 | 46.3 | 46.3 | 46.3 | ........ | ........ |
| Steam pressure (gage) | 164 | 216 | 205.1 | 205.3 | 204.7 | 195.7 | 197.3 | 197.0 | 202 | 201.4 |
| Quality or superheat per cent or degree | 95.4% | 98.9% | 62.9° | 56.4° | 55.1° | 71.4° | 99.4% | 99.15% | 100% | 100% |
| Temperature of feed water | 141 | 188 | 64.7 | 64.0 | 63.8 | 60.0 | 206 | 67 | 211.6 | 211.4 |
| Draft between furnace and ashpit | ........ | .23 | .34 | .23 | .20 | .34 | ........ | ........ | ........ | ........ |
| Draft between ashpit and base of stack | Nat. | Howd | .67 | .44 | .35 | 1.19 | .78 | .85 | .31 | .94 |
| Temperature at base of stack | ........ | 466 | 495 | 474 | 461 | 535 | 513 | 591 | 491 | 545 |
| Temperature in boiler room | ........ | ........ | ........ | ........ | ........ | ........ | 57 | 67 | 94 | 98 |
| Temperature external air | ........ | ........ | ........ | ........ | ........ | ........ | 38 | 53 | ........ | ........ |
| Kind of coal | ........ | ........ | Run of mine | | | | ........ | ........ | Pocahontas best | |
| Calorific value (dry coal) b.t.u. | 14,420 | 14,470 | 13,864 | 14,256 | 14,251 | 14,135 | 14,213 | 14,342 | 15,838 | ........ |
| Per cent moisture | ........ | ........ | 2.35 | 2.51 | 2.62 | 3.46 | 1.54 | 2.82 | ........ | ........ |
| Ash and refuse (based on dry coal) per cent | 19.8 | 8.89 | 11.57 | 12.15 | 11.00 | 10.41 | 10.15 | 10.20 | 6.00 | 3.83 |
| Coal fired per hour (lbs.) | 2,170 | 1,188 | 1,687 | 1,499 | 1,265 | 2,068 | 1,444 | 1,614 | ........ | ........ |
| Dry coal per hour (lbs.) | ........ | ........ | 1,647 | 1,461 | 1,232 | 1,997 | 1,412 | 1,568 | 873 | 1,425 |
| Water evaporated per hour | 17,000 | 11,730 | 14,195 | 12,481 | 10,917 | 16,688 | 14,042 | 14,588 | 9,480 | 15,725 |
| Coal per sq. ft. g.s. per hour | 15.6 | 20.1 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Dry coal per sq. ft. g.s. per hour | ........ | ........ | 25.0 | 22.2 | 18 7 | 30.3 | 21.3 | 20.2 | 15.1 | 24.6 |
| Dry coal per sq. ft. h.s. per hour | ........ | ........ | .54 | .48 | .405 | .655 | .57 | .62 | .34 | .554 |
| Actual evaporation lbs. water per lb. coal | 7.84 | 9.89 | 8.42 | 8.33 | 8.63 | 8.07 | ........ | ........ | ........ | ........ |
| Actual evaporation lbs. water per lb. dry coal | ........ | ........ | 8.62 | 8.54 | 8.86 | 8.36 | 9.90 | 9.23 | 10.76 | 10.95 |
| Equivalent evaporation from and at 212° per lb. coal | 8.84 | 10.55 | 10.47 | 10.33 | 10.70 | 10.10 | ........ | ........ | ........ | ........ |
| Equivalent evaporation from and at 212° per lb. dry coal | ........ | ........ | 10.72 | 10.60 | 10.99 | 10.47 | 10.40 | 11.08 | 11.42 | 11.61 |
| Actual evaporation per sq. ft. h.s. per hour | 4.9 | 5.0 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Equivalent evaporation from and at 212° per sq. ft. h.s. per hour | 5.52 | 5.35 | 5.78 | 5.30 | 4.44 | 6.85 | 5.90 | 6.95 | 3.88 | 6.43 |
| Efficiency based on dry coal | 59.2[3] | 70.5[3] | 75.03 | 72.15 | 74.83 | 71.88 | 71.00 | 75.0 | 74.4[4] | 74.0[4] |
| Excess air (dry coal basis) per cent | ........ | ........ | 63.19 | 60.36 | 64.73 | 74.30 | ........ | ........ | ........ | ........ |
| Firing | ........ | ........ | Hand fired | | | | Mech. stoker | | Hand fired | |
| Factor of evaporation | 1.126 | 1.081 | 1.244 | 1.241 | 1.240 | 1.252 | 1.055 | 1.20 | ........ | ........ |

[1] S.S. "El Sud."

[2] S.S. "John D. Rockefeller."

[3] Based on coal "as fired."

[4] Based on combustible.

TABLE VI.—PERFORMANCE OF OIL FIRED BOILERS

| Make | Ward | Foster | Foster | Foster | B & W 2-in. tubes | B & W 2-in. tubes | B & W 2-in. tubes | White Forster | White Forster | Yarrow |
|---|---|---|---|---|---|---|---|---|---|---|
| Reference | S.N.E. 1916 | Foster catalog | Foster catalog | Foster catalog | "Marine Steam" | "Marine Steam" | "Marine Steam" | S.NA&ME 1920 | S.NA&ME 1920 | I.N.A. 1912 |
| Heating surface, sq. ft. | 4,405 | 3,050 | 3,050 | 3,050 | 4,000 | 4,000 | 2,571 | 7,565 | 7,565 | 5,435 |
| Furnace volume, cu. ft. | 476 | ........ | ........ | ........ | 445 | 445 | 217 | 751 | 751 | ........ |
| Superheating surface | 0 | 338 | 338 | 338 | 0 | 0 | 0 | 753 | 753 | 1,265 |
| Steam pressure (gage) | 194.75 | 190.4 | 212.2 | 207.5 | 300 | 302 | 210.7 | 295.6 | 296.0 | 243.7 |
| Quality or superheat | .994 | 56.6 | 67.8 | 66.6 | .985 | .979 | .999 | 80.8° | 67.4° | 82.5° |
| Temperature of feed water °F | 183.6 | 81.0 | 64.1 | 65.0 | 201.3 | 200.6 | .......... | 198.0 | 194.7 | 63.5 |
| Kind of draft | C.S. | Forced | Forced | Forced | C.S. | C.S. | C.S. | C.S. | C.S. | C.S. |
| Draft inches of water | 2.34 | .68 | .90 | 1.00 | 3.26 | 5.11 | 1.64 | 9.54 | 4.95 | 2.44 |
| Temperature in boiler room | 62 | ........ | ........ | ........ | 94.3 | 96.9 | 79 | 109.1 | 115.5 | ........ |
| Temperature base of stack | 418 | 567 | 646 | 718 | 620 | 720 | 447 | 729 | 617 | 685 |
| Calorific value, b.t.u. | 19,879 | 19,231 | 19,311 | 19,311 | 19,525 | 19,525 | 17,812 | 19,300 | 19,380 | ........ |
| Oil burned per hour | 1,509.7 | 1,329 | 1,642 | 1,985 | 2,532 | 4,005 | 1,202 | 11,355 | 6,390 | 5,695 |
| Water evaporated per hour | 23,532 | 16,320 | 19,621 | 22,980 | 37,504 | 60,325 | .......... | 153,467 | 92,208 | 68,387 |
| Oil burned per sq. ft. h.s. per hour | .343 | .436 | .538 | .651 | .633 | 1.001 | .467 | 1.50 | .845 | .85 |
| Actual evaporation, lbs. water per lb. oil | 15.59 | 12.28 | 11.95 | 11.58 | 14,812 | 15.06 | 14.83 | 13.52 | 14.43 | 12.00 |
| Equivalent evap. from & at 212° per lb. oil | 16.73 | 14.99 | 14.91 | 14.42 | 15.62 | 15.82 | 15.72 | 15.15 | 16.17 | 15.2 |
| Equivalent evap. from & at 212° per sq. ft. h.s. | 5.74 | 6.53 | 8.03 | 9.39 | 9.89 | 15.84 | 7.35 | 22.73 | 13.66 | 15.9 |
| Excess air, per cent | ......... | 43.03 | 24.88 | 22.08 | 11.3 | 8.3 | .......... | 18.0 | 24.4 | ........ |
| Efficiency, per cent | 81.68 | 75.64 | 74.92 | 72.46 | 77.64 | 78.64 | 79.5 | 76.15 | 81.37 | ........ |
| Type and number of burners | 7 Bur.S.E. | 4 Dahl | 4 Dahl | 4 Dahl | 8 | 13 | 11 Peabody | Peabody | Peabody | ........ |
| Oil per burner per hour | 216 | 332 | 411 | 496 | 316 | 308 | 300 | 1,032.3 | 710 | ........ |
| Oil pressure at burner | 258 | 149.2 | 136.0 | 232.1 | 196 | 193 | 175.6 | 245.9 | 170.6 | ........ |
| Oil temperature at burner | 163 | 227.3 | 60.0 | 60.0 | 180 | 181 | 184.0 | 120.0 | 120.0 | ........ |
| Factor of evaporation | 1.074 | 1.221 | 1.248 | 1.249 | 1.055 | 1.050 | 1.052 | 1.1206 | 1.1204 | ........ |

# CHAPTER V

## COMBUSTION

### Part I—Combustion of Coal

**45. Chemistry of Combustion.**—Combustion is a rapid chemical union of fuel and the oxygen in the air. Fuel must first be brought to the ignition temperature by the application of heat. At the ignition temperature the hydrocarbons are driven off and ignited; the remaining fixed carbon unites with the oxygen of the air forming carbon dioxide ($CO_2$). The fuel absorbs heat during the first stage of combustion, and burns and gives up heat during the second and third stages.

During the process of burning, the proper amount of oxygen must be brought in contact with each particle of the fuel so that complete combustion of the fuel will result and $CO_2$ be formed. The exact chemical combinations of the hydrocarbons and elements in the fuel are rather complicated and a complete analysis of the reactions taking place during combustion is outside of the scope of this book. At the completion of the combustion the carbon has united with oxygen forming $CO_2$ and the hydrogen has united with oxygen forming $H_2O$. Thus

$$C+O_2=CO_2$$
$$H_2+\tfrac{1}{2}O_2=H_2O$$

The elements unite in proportion to their atomic weights as follows:

$$12 \text{ lbs. } C+(2\times16) \text{ lbs. } O=44 \text{ lbs. } CO_2$$
$$(2\times1) \text{ lbs. } H+16 \text{ lbs. } O=18 \text{ lbs. } H_2O$$

Dividing the above by 12 and 2 we have,

$$C+2\ 2/3\ O=3\ 2/3\ CO_2$$
$$H+8O=9H_2O$$

Thus each pound of carbon unites with 2 2/3 lbs. of oxygen form-

ing 3 2/3 lbs. of carbon dioxide and each pound of hydrogen unites with 8 lbs. of oxygen forming 9 lbs. of water.

If sufficient air is not supplied to give enough oxygen to unite with the carbon and form $CO_2$, carbon monoxide (CO) is formed. If $CO_2$ is formed the combustion is said to be "complete," if CO is formed the combustion is said to be "incomplete." In the latter case the reaction is,

$$12C + 16O = 28CO$$

or,

$$C + 1\ 1/3\ O = 2\ 1/3\ CO$$

Thus, if only 1 1/3 lbs. of oxygen are supplied per pound of carbon the carbon unites with the oxygen forming 2 1/3 lbs. of CO.

During the above reactions heat is liberated as follows:

| | |
|---|---|
| $C + O_2 = CO_2$ | 14,600 b.t.u. per lb. C |
| $C + O = CO$ | 4,500 b.t.u. per lb. C |
| $H_2 + O = H_2O$ | 61,950 b.t.u. per lb. H. |

If the CO is burned to $CO_2$ by the application of more oxygen we have,

$$2CO + O_2 = 2CO_2 \qquad 10,100 \text{ b.t.u. per lb. C.}$$

In the total process of combustion of C to CO and thence to $CO_2$ we have $4,500 + 10,100 = 14,600$ b.t.u. or the same heat liberated when carbon is burned to $CO_2$ directly. The above clearly shows the great loss of the heat in the fuel if sufficient oxygen is not supplied for complete combustion. When carbon monoxide (CO) is formed only 4,500 b.t.u. are liberated per pound of carbon against 14,600 b.t.u. when carbon dioxide ($CO_2$) is formed.

Reference to Table II will show that the amount of hydrogen in coal is small, seldom running over 5 per cent. As a rule a large proportion of this is already in combination with oxygen in the form of water ($H_2O$). Hence the amount of oxygen required for the hydrogen and the heat liberated by its union with oxygen has a very small influence on the combustion.

**46. Air Required for Combustion.**—For engineering purposes it is customary to assume that all the coal consists of carbon and

2 2/3 lbs. of air are required per pound for complete combustion. The oxygen for combustion is supplied from atmospheric air which consists of 23 parts oxygen to 77 parts nitrogen by weight. The ratio of air to oxygen is $1{,}000/23 = 4.35$. Therefore $4.35 \times 2.66 = 11.6$ lbs. of air must be supplied per pound of fuel for complete combustion. For atmospheric air at 60°F and 14.7 lbs./sq. in. this is equivalent to $11.6/.076 = 152$ cu. ft. of free air. The nitrogen in the air takes no part in the combustion process and merely dilutes the uptake gases. The actual theoretical amount of air required, taking account of the hydrogen and sulphur in the fuel, is.

$$\text{Pounds of air per pound of coal} = 11.6C + 34.2(H - O/8) + 4.35S.$$

**47. Excess Air.**—In the foregoing article it was assumed that the fuel was pure carbon and that perfect and complete combustion was taking place, the exact amount of oxygen being supplied to each particle of fuel at the proper time. In the actual combustion of coal in a furnace these conditions are impossible. If the correct amount of air were supplied to burn one pound of carbon, there would not be sufficient air to burn the hydrogen, which requires three times as much per pound as carbon. Also, due to the thickness and unevenness of the fuel bed and unequal distribution of the entering air, each particle of fuel would not receive its full amount of oxygen at the proper time. Under these conditions, part of the combustion would be complete and a large part incomplete and carbon monoxide would be formed with its resultant loss of 10,100 b.t.u. per pound of fuel.

To obviate a deficiency of air at any point in the fuel bed it is necessary to supply 20 to 100 per cent more air than the theoretical amount. This increase is called excess air. The actual amount necessary to insure complete combustion varies with the kind of fuel and the care used in firing. As all the excess air has to be heated and as a large part of this heat escapes up the stack and cannot be recovered, the amount of air used should be kept as low as possible. The loss caused by incomplete combustion is much greater than that caused by the dilution of the furnace gases; hence any error should always be towards too great excess air rather than too little.

The amount of oxygen in the air is 21 per cent by volume, and

as the volume of $CO_2$ formed is the same as the oxygen used, it follows that 21 per cent of the volume of the flue gases should be $CO_2$ if the theoretical amount of air is supplied and perfect combustion results. With an excess supply of air, the percentage of $CO_2$ in the uptake gases decreases, being 10.5 per cent when the excess air is 100 per cent. The percentage of $CO_2$ in the uptake gases thus is an indication of the amount of excess air supplied. For good operation without the formation of carbon monoxide the percentage of $CO_2$ should be between 9 and 16 per cent. Values lower than 9 per cent generally indicate too large amount of air and values higher than 16 per cent are generally an indication that CO exists in the uptake gases. Instruments known as $CO_2$ recorders are on the market which give a daily graphical record of the variation of the volume of $CO_2$ in the flue gases. These thus serve as a guide for the firemen in adjusting their air supply and give a permanent record for the chief engineer and superintending engineer of the boiler performance.

**48. Method of Supplying Air.**—If all the air were supplied from below the fuel bed, carbon dioxide ($CO_2$) would be formed in the lower part of the bed and then later this $CO_2$ would be reduced to carbon monoxide (CO) as it came in contact with the incandescent carbon at the top of the fuel bed. Moreover, as the hydrocarbons were distilled off from the top of the fire there would be little or no oxygen present to unite with these gases and liberate heat by the formation of $CO_2$ and $H_2O$. Under these conditions there would be a large loss of available carbon in the form of CO and also smoke would be produced.

To secure proper combustion it is necessary that a sufficient amount of air be distributed over the top of the fuel bed. This air must be brought to a high temperature before coming into contact with the volatile gases, otherwise the gases will be chilled and smoke result. It is also very essential that air supplied at the top of the fuel bed should be thoroughly mixed with the gases so that complete oxidation will take place. This air supply for the top of the fire is sometimes passed up through the fuel bed, but more often it is admitted through openings in the firedoor or at the bridge wall in the rear. The Wager bridge wall shown in Figs. 26 and 28 is an excellent means of admitting air to the top of the fire and as the air comes in through

small slots proper mixing will take place. The heating of the air can be accomplished by means of the Howden heater (Figs. 32 and 11) or by passing the air and gases over incandescent portions of the fire. Air is especially needed at the top of the fire after firing when the volatile gases are being distilled off; hence the practice of covering part of the fire at one time with fresh coal leaves the incandescent part of the fuel bed to assist in the combustion of the gases given off from the fresh coal.

**49. Losses in Burning Coal.**—When coal is burned in the furnace of a boiler there are various losses of heat, some avoidable and some unavoidable. The result is, that only between 60 and 85 per cent of the heat liberated during combustion is absorbed

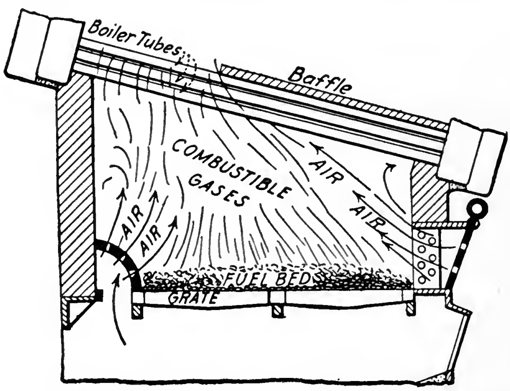



Fig. 26.—Wager bridge wall fitted with water-tube boiler.

by the water in the boiler. The various losses in burning coal can be divided up as follows:

1. Combustible in the ash.
2. Moisture in the fuel.
3. Superheating the moisture produced by hydrogen in fuel.
4. Incomplete and imperfect combustion.
5. Heat carried away by stack gases ("dry chimney gases").
6. Superheating of the moisture in air.
7. Radiation.
8. Unburned carbon in the form of smoke.

*Moisture in Fuel.*—The economic loss and the increase bunker space caused by moisture in the coal have already been mentioned. Here we are only concerned with the loss caused during combustion. All the moisture contained in the coal requires heat to convert it into superheated steam at furnace temperature. As

the gases are cooled in their passage over the heating surface part of the heat of superheat is given up; but all the heat in this superheated steam above the uptake temperature is carried up the stack and is lost. This loss may be expressed in b.t.u. per hour as follows:

$$(q_1+r_1-q_2+ct_s)w$$

where $q_1 =$ heat of liquid at atmospheric pressure.
$r_1 =$ heat of vaporization at atmospheric pressure.
$q_2 =$ heat of liquid at temperature of fuel before firing.
$c =$ specific heat of superheated steam.
$t_s =$ degree of superheat in uptake (uptake temperature $-$ 212°).
$w =$ lbs. of water fired with coal per hour.

*Moisture in the Air.*—This loss is due to converting the moisture in the air from water at boiler room temperature into superheated steam at stack temperature. This loss is of no importance on clear days but increases on damp and foggy days. If the humidity and quantity of air supplied are known the loss can be easily computed by the method used in the preceding article.

*Hydrogen in Fuel.*—This is a loss similar to the two preceding ones. The hydrogen in the fuel unites with oxygen in the air forming water. This water absorbs heat when it is converted into superheated steam and all the heat contained in this steam above uptake temperature is lost up the stack. The loss in b.t.u. per hour is expressed as follows: $9(q_1+r_1-q_2+ct_s) \times$ coal fired per hour $\times$ per cent hydrogen in fuel.

*Incomplete Combustion.*—This loss already has been dealt with at length. It is caused by too small an amount or the improper distribution of the air supply resulting in the formation of carbon monoxide (CO) instead of carbon dioxide ($CO_2$). The percentage of $CO_2$ in the flue gases is an approximate measure of this loss but the exact loss can only be ascertained by a flue gas analysis in which the percentage of CO is determined. The loss in b.t.u. per pound of fuel due to incomplete combustion is

$$=10{,}100\mathrm{C} \times \frac{\mathrm{CO}}{\mathrm{CO}+\mathrm{CO_2}}$$

where $C$ = weight percentage of carbon in fuel actually burned.
$CO$ = volume percentage of $CO$ in flue gas.
$CO_2$ = volume percentage of $CO_2$ in flue gas.

This loss can be practically eliminated by careful attention to firing, allowing sufficient excess air, and the proper mixture of the air and combustible gases.

**Dry Stack Gases.**—This loss is always the largest one except for poorly operated boilers and is one that is unavoidable. It is caused by the heat carried away by the stack gases. The temperature of the gases leaving the boiler must always be higher than that of the steam in the boiler or heat will pass from the steam to the gases. It is common practice to keep this temperature about 100° higher than that of the steam. In order to keep this loss as low as possible, the excess air should be kept down within reasonable limits but not so low that incomplete combustion will result. The loss due to stack gases increases when the boiler is forced, and is generally a large loss for boilers in destroyers and other high speed craft when operating at maximum capacity, due to the high temperature of the stack gases. A considerable portion of the heat in these waste gases can be recovered by the use of Howden air heaters (Art. 66), waste type superheaters (Art. 79), and economizers (Art. 182). The loss in b.t.u. per lb. of fuel is,

$$= W.c(t_f - t_a)$$

where $W$ = the weight of flue gases per pound of fuel.
$c$ = specific heat of flue gases.
$t_f$ = temperature of gases leaving boiler.
$t_a$ = temperature of air entering furnace.

By careful firing the amount of excess air can be reduced and the boiler efficiency increased. With mechanical stokers (Art. 53), pulverized coal (Art. 54), and liquid fuel, because of the better mixing of the fuel and air, the amount of excess air required is much less than with intermittent and irregular hand firing with coal. This accounts for the higher boiler efficiency obtained with the methods of combustion and the fuels mentioned.

**Radiation.**—The loss due to radiation is generally in the neighborhood of 5 per cent and can be reduced by insulation of the boiler and furnace. Water-tube boilers should have an ade-

quate brick furnace lining to reduce radiation as much as possible. In high speed craft this furnace brickwork has been greatly reduced to save weight, but on merchant ships heavy insulation consisting of two or more thicknesses of brick should be used (see Fig. 15). Scotch boilers have no furnace radiation, but the radiation from the shell is large unless carefully lagged.

**50. Heat Balance.**—In all complete boiler tests the various losses mentioned above are calculated and tabulated. These losses together with the heat absorbed by the boiler constitute what is known as a heat balance. The heat absorbed by the water is calculated from the evaporation and the losses 1 to 6 in Art. 49 are then computed. The difference between the heat value of the coal and the heat absorbed plus losses 1 to 6 is tabulated as "radiation and unaccounted for." A few typical heat balances are given in Table VIII.

TABLE VIII.—HEAT BALANCES

| | Foster W.T. with coal[1] | White Forster with oil[2] | B&W with oil[3] | Average operation coal[4] |
|---|---|---|---|---|
| Rate of combustion | 19.1[6] | 1.50[5] | 0.633[5] | ...... |
| Heat absorbed by boiler, per cent | 72.87 | 76.15 | 77.64 | 68.58 |
| Loss due to combustible in ash, per cent | 2.40 | ....... | ....... | 4.75 |
| Loss due to moisture in fuel, per cent | 0.25 | ....... | | 0.35 |
| Loss due to hydrogen in fuel, per cent | 3.56 | 7.12 | 7.64 | 3.26 |
| Loss due to moisture in air, per cent | 0.19 | 0.22 | ....... | ....... |
| Loss due to incomplete combustion, per cent | 2.62 | 0.31 | ....... | 1.78 |
| Loss due to dry stack gases, per cent | 11.52 | 12.71 | 9.51 | 16.37 |
| Radiation and unaccounted for, per cent | 6.59 | 2.23 | 5.21 | 4.91 |
| Basis of balance | dry coal | ....... | ....... | dry coal |
| Temp. of gases at base of stack | ....... | 729° | 620° | ....... |
| Excess air, per cent | 60.87 | 19.5 | 11.3 | ....... |

[1] Foster catalog.
[2] E. H. Peabody S.N.A. & M.E. 1920—U. S. Fuel Oil Testing Plant.
[3] B&W Boiler at U. S. Fuel Oil Testing Plant.
[4] Mark's Handbook "average operating conditions."
[5] Oil per sq. ft. H.S.
[6] Dry coal per sq. ft. G.S.

**51. Boiler Efficiency.**—Boiler efficiency has already been defined in Art. 41. It should be observed here, however, that the overall efficiency of the boiler and furnace which is the one usually used, takes account of all the losses given in the preceding article. The percentage of heat absorbed by the boiler in the above table is the boiler efficiency as defined in Art. 41.

**52. Temperature of Gases in Boiler.**—The theoretical maximum temperature in the furnace can be easily calculated from the calorific value of the fuel and the weight and specific heat

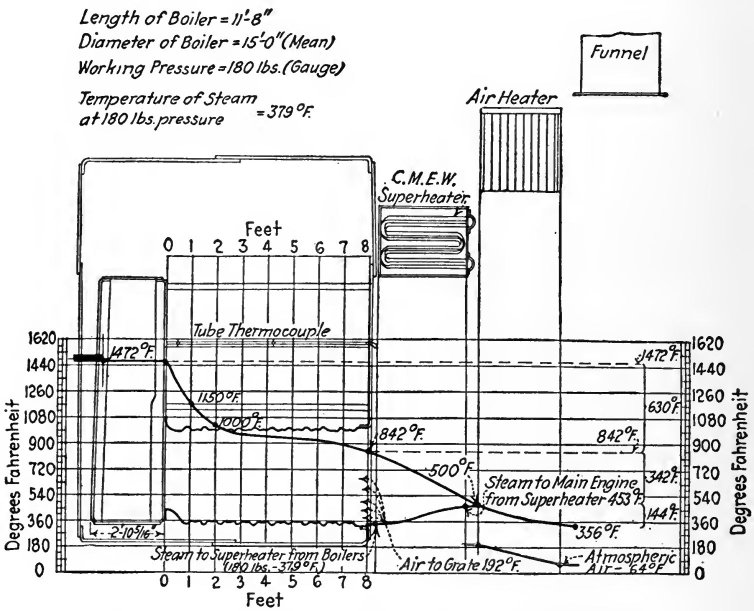



FIG. 27*a*.—Variations in temperature in a Scotch boiler.

of the flue gases. The actual furnace temperature for coal varies roughly between 2,000° and 3,000°F. depending on the amount of excess air; for oil fuel the temperatures are higher, generally being in the neighborhood of 3,000°. Figures 27a and 27b show the variation of the temperature of the gases during their passage through the boiler and are worthy of careful study. These figures are of special value in studying the temperature when superheaters are installed. Figure 27b is for a Yarrow express type water-tube boiler taken from a paper by Alfred Yarrow before the Institution of Naval Architects, 1912; and

Fig. 27a is for a Scotch boiler and is taken from a paper by Maurice S. Gibbs before the North East Coast Inst. of Engineers and Shipbuilders, 1920.

**53. Mechanical Stokers.**—The mechanical stoker so far has been practically untried on shipboard, although it is giving excel-

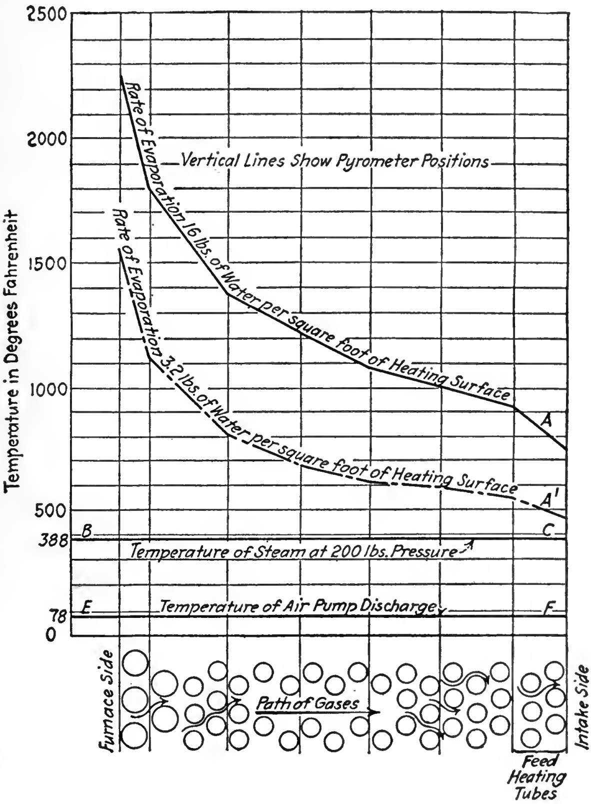



FIG. 27*b*.—Variations in temperature in Yarrow boiler.

lent service in power plants on shore and is almost universally used in all large plants using coal. The efficiency of a boiler fitted with a stoker is nearly the same as when using fuel oil. The steam necessary to operate the stoker would not greatly exceed the 1 1/2 per cent required by a fuel oil installation. Further, the labor expense certainly would be nearly as low as that for fuel oil and the duties of the firemen will be as easy as

with oil. Stokers will involve some increased weight and space and the boilers will have to be raised to allow for the installation of stoker. These factors, however, should cause no serious difficulties on merchant ships. The increased weight and space required over hand fired boilers will be offset to some extent by the smaller boilers required because of the higher rates of combustion possible with stokers. The boiler will probably not be raised more than 6 or 8 ft. and any loss in metacentric height can be made up by an increase in beam. In fact, because of the better design that can be produced with increased beam it may even be possible to stow the coal in raised bunkers so that it can be delivered to the stoker hoppers by gravity.

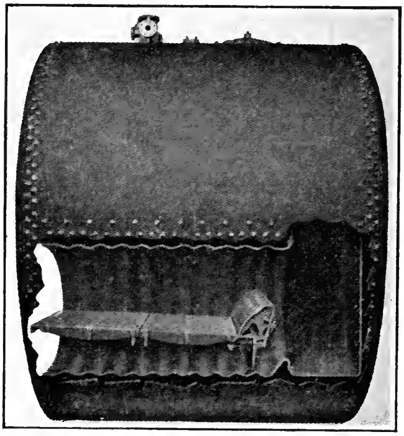


Fig. 28.—Wager bridge wall fitted in Scotch boiler.

With improved bunkering facilities such as the Michener bunkering machine, much of the dirt and expense of bunkering could be eliminated. Mechanical conveyors could be installed for handling coal from the bunkers to the stoker and mechanical bunker trimmers could also be used for trimming the coal in the bunkers. It is interesting to note that the methods of handling coal today on shipboard are practically the same that were used on the first coal burning steamship of 70 years ago.

Comparative tests have been made on shore with the Emergency Fleet Corporation's four pass water-tube boiler, both hand fired and with underfeed stokers (Proceedings A. S. M. E., 1919). At a rate of combustion of 20 lbs. dry coal per square foot of grate surface, the efficiency based on dry coal was 71 per cent for the hand fired boiler and 75 per cent when fitted with mechanical stokers. The hand fired boiler in these tests had an added advantage by being fitted with Wager bridge walls.

The advantages of mechanical stokers over hand firing can be summarized as follows:

1. Continuous and uniform firing and uniform distillation of hydrocarbons.
2. Uniform thickness of fire; hence uniform draft and no holes in fuel bed.
3. No chilling of volatile gases or dilution by frequent opening of furnace doors.
4. Higher rates of combustion.
5. Reduced fireroom force; less arduous duties for firemen.
6. Higher boiler efficiency (less excess air required).
7. Personal element of firing eliminated.
8. Smokeless combustion.

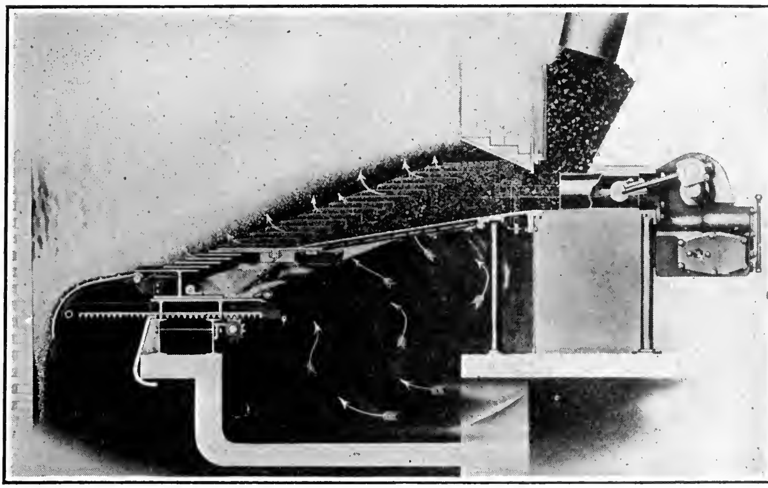


FIG. 29.—Riley self-dumping underfeed stoker

*Types of Stokers.*—A large variety of mechanical stokers are in existence. The main types are (1) chain grate, (2) side feed, (3) step grates and (4) underfeed. A Riley underfeed stoker is shown in Fig. 29, an arrangement of a Riley stoker with a B & W marine boiler in Fig. 29a; and a Jones underfeed stoker with a stationary internally fired boiler in Fig. 29b.

**54. Pulverized Coal.**—Pulverized coal is finding a large field in stationary plants and it probably will be introduced on shipboard if compact crushing and pulverizing apparatus can be devised. The advantages of pulverized fuel are: cheapness; ease of burning and reduced fireroom force; boiler efficiency as high as with liquid fuel because of excellent combustion and less amount of excess air required; poor grades of coal can be used;

no ashes to handle; no stand-by losses; high rates of combustion. The cost of crushing, drying, pulverizing and delivering to

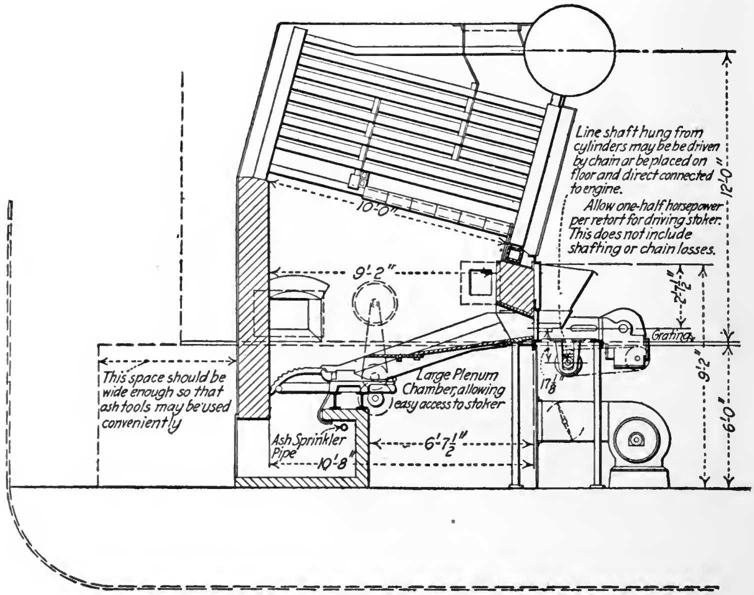



Fig. 29a.—Riley underfeed stoker fitted with a marine water-tube boiler.

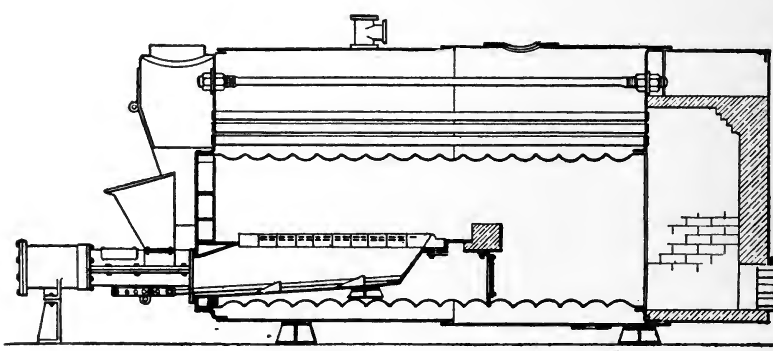


Fig. 29b.—Jones stoker fitted in internally fired boiler.

boilers is given by Scheffler and Barnhurst (A. S. M. E., 1919) as 18 kw-hours per ton of coal. Allowing 1 1/2 lbs. of coal per horsepower per hour, this is equivalent to 1 1/2 per cent of

the boiler output which is practically the same as required for fuel oil and stokers. Figure 30 shows a boiler fitted for burning

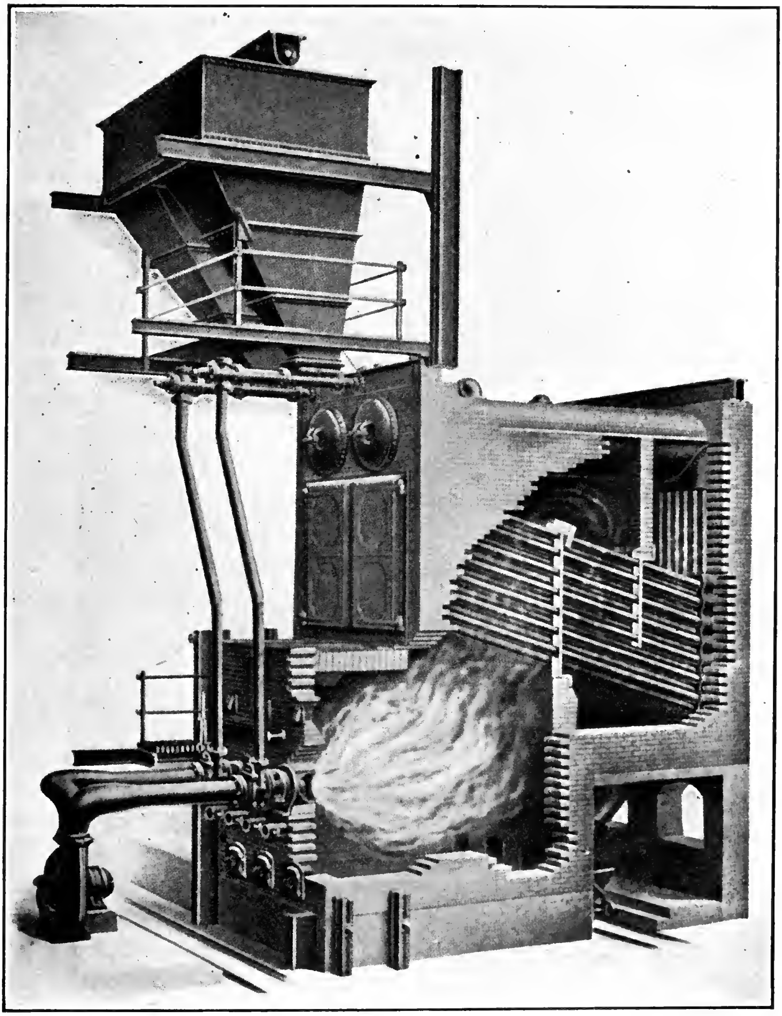


FIG. 30.—Burning pulverized coal with Fuller Engineering Co.'s equipment in stationary boiler.

pulverized fuel with apparatus manufactured by the Fuller Engineering Co. of Allentown, Pa.

## Part II—Combustion of Fuel Oil

**55. General.**—Oil is burned under boilers by injecting the oil into the furnace in the form of a fine atomized spray (Fig. 31). The oil is pumped from the fuel tanks forward and aft to a fuel tank amidships by the fuel oil transfer pump (booster pump). The fuel oil service pump takes the oil from the tank located amidship and delivers it to the burners under a pressure of between 100 and 250 lbs. per square inch. The oil is passed through a heater on the way to the burners where it is raised to a high temperature in order to reduce its viscosity so that it can be easily atomized into a fine spray by the burner. The air for combustion is admitted through openings (registers) around the burner at high velocity and mixes thoroughly with the atomized oil in the furnace, resulting in complete oxidation of the fuel and the liberation of heat.

**56. Fuel Oil Burners and Mechanical Atomization.**—In mechanical atomization the oil is atomized by the action of centrifugal force. The oil is given a whirling motion in the burner either by means of a helical groove or passage, or by tangential holes. The oil is delivered to the burner under high pressure and the rotary motion imparted by the burner breaks the oil up into a fog-like spray. The oil leaves the burner in a thin conical sheet (Fig. 31) and at once mingles with the air which enters around the burner at high velocity. Figure 31 shows an illustration of the Schutte-Koerting burner and register. Heating of the oil is essential to reduce its viscosity so that the burner can perform its function; the temperature to which the oil must be heated varies with the viscosity of the oil and ranges between 100 and 250°. For successful atomization, the oil should be heated so that its viscosity is between 10 and 3° Engler scale, depending on the flash point of the oil.

The capacity of a given burner depends upon the pressure of the oil—increasing the pressure increases the capacity. The maximum capacity of burners used in the merchant service is about 600 lbs. of oil per hour while 1,500 lbs. per hour and even

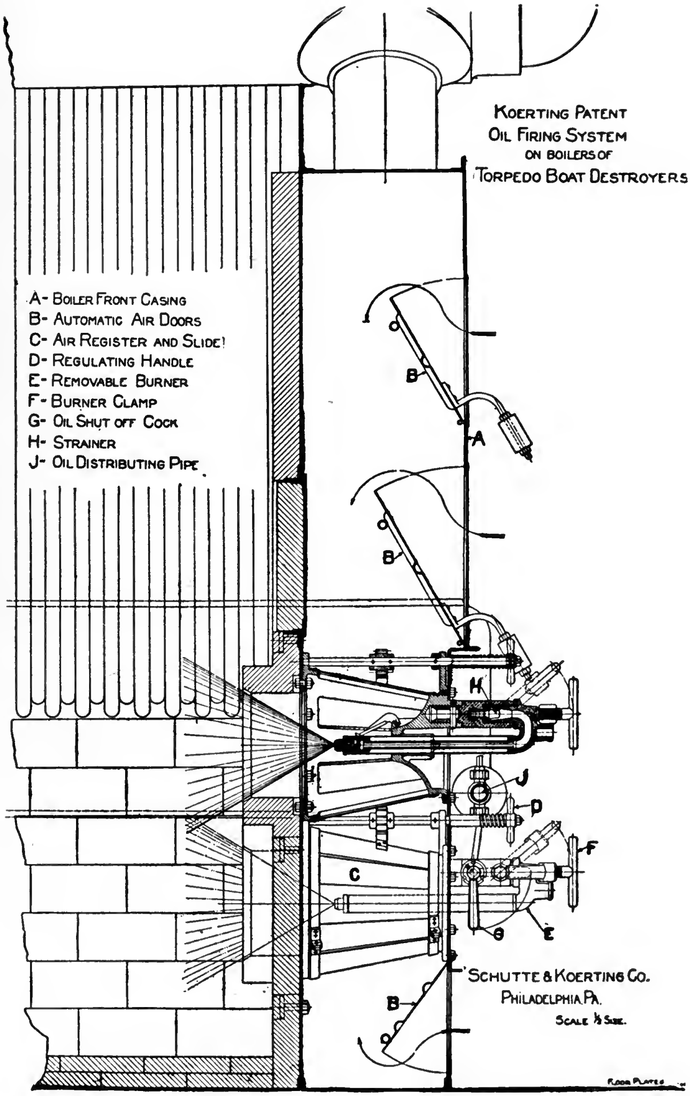

KOERTING PATENT
OIL FIRING SYSTEM
ON BOILERS OF
TORPEDO BOAT DESTROYERS.
A- BOILER FRONT CASING
B- AUTOMATIC AIR DOORS
C- AIR REGISTER AND SLIDE
D- REGULATING HANDLE
E- REMOVABLE BURNER
F- BURNER CLAMP
G- OIL SHUT OFF COCK
H- STRAINER
J- OIL DISTRIBUTING PIPE
A
B
C
D
E
F
G
H
J
SCHUTTE & KOERTING CO.
PHILADELPHIA, PA.
SCALE ½ SIZE.

FIG. 31.

one ton of oil per hour has been attained during tests with naval boilers at the U. S. Fuel Oil Testing Plant.[1]

The capacity of a boiler is increased both by increasing the oil pressure at the individual burners and by adding additional burners.

In stationary installations steam and compressed air are used for atomizing the oil in the burners. The steam atomizer is most universally used and gives excellent results. However, because of the steam required for operation and the necessity of make up feed to replace this steam used, steam atomizers are not used on shipboard. The mechanical atomizer described

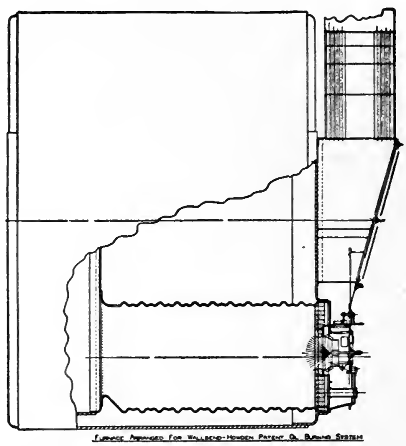


FIG. 32.—Arrangement of Howden system of forced draft on Scotch boiler. (Oil.)

above has been developed in marine work and while perhaps slightly less efficient than the steam atomizer, it gives excellent results and entire satisfactory performance.

**57. The Babcock & Wilcox Mechanical Atomizer.**—Figure 33 shows a drawing of the B & W atomizer.

A conical cast iron casting, 1, is placed in the front wall of the boiler setting as an orifice through which the air and oil are admitted to the furnace. The main register casting, 2, is fitted with automatic doors, 3, by which the quantity of air supplied to the burners may be regulated or shut off entirely. These doors are so designed that they will close automatically in case of a flare-back in the furnace or the bursting of a boiler

[1] E. H. Peabody, "Recent Advance in Oil Burning." *Trans. Soc. N. A. & M. E.*, 1921.

tube, thus protecting the fireman who may be standing in front of the boiler. A cover plate, 4, is attached to the front of the register casting, and a radiation guard, 5, is held between them. The operation of the automatic air doors, 3, is controlled by four cams, 6, forming part of a spider casting which can be rotated by the handle 7. By moving the handle to the extreme right of the slot through which it slides (the position shown in Fig. 33), all the air doors are closed; by moving it to the left the doors are gradually opened.

Passing through the center opening in the cover plate, 4, is a hollow cylindrical distance piece having at its outer end a quick detachable coupling, 9, and yoke, 10. To the other end of this distance piece is

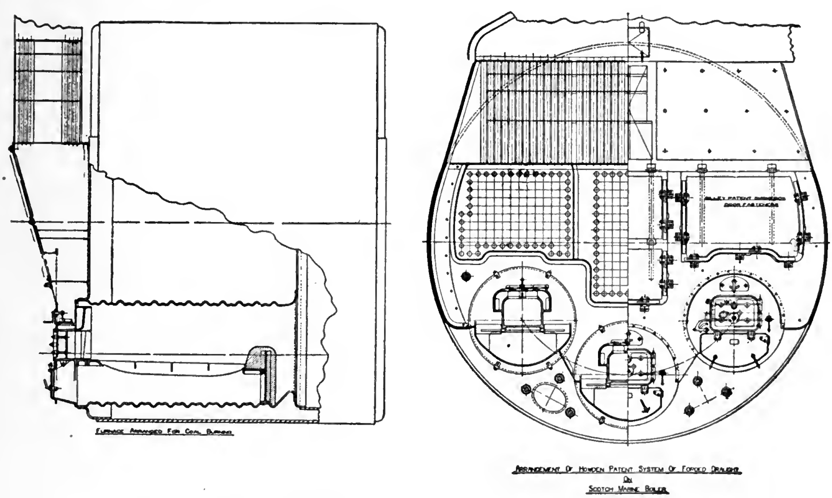


Fig. 32—Arrangement of Howden system of forced draft on Scotch boiler. (Coal).

fastened an aluminized-steel conical shaped impeller plate, 11, for regulating the distribution of air at the nozzle of the burner. Passing through the distance piece is the mechanical atomizer, 13, which is held in place and connected to the fuel oil supply lines through the coupling, 9, and yoke, 10, thus making the atomizer, 13, the distance piece and the center impeller, 11, a rigid unit when in operation. The distance piece is so designed that it may be moved along its axis, thus moving the impeller plate, 11, in and out with reference to the cone, 1, and decreasing or enlarging at will the clear area for the passage of air around the outside of the impeller plate. The distance piece is held in place by a set screw, 12.

The atomizer ends in a burner which imparts a rotary motion to the oil as it emerges in a conical spray. The burner ejects this spray through a central opening in the impeller plate, 11, which plate has blades like

those of a fan to give a rotary motion to the air as it enters the furnace around the burner tip. The cone, 1, also has blades which give a whirling motion to the air entering around the edge of the impeller plate, this air being directed along the axial line of the burner by the walls of the trun-

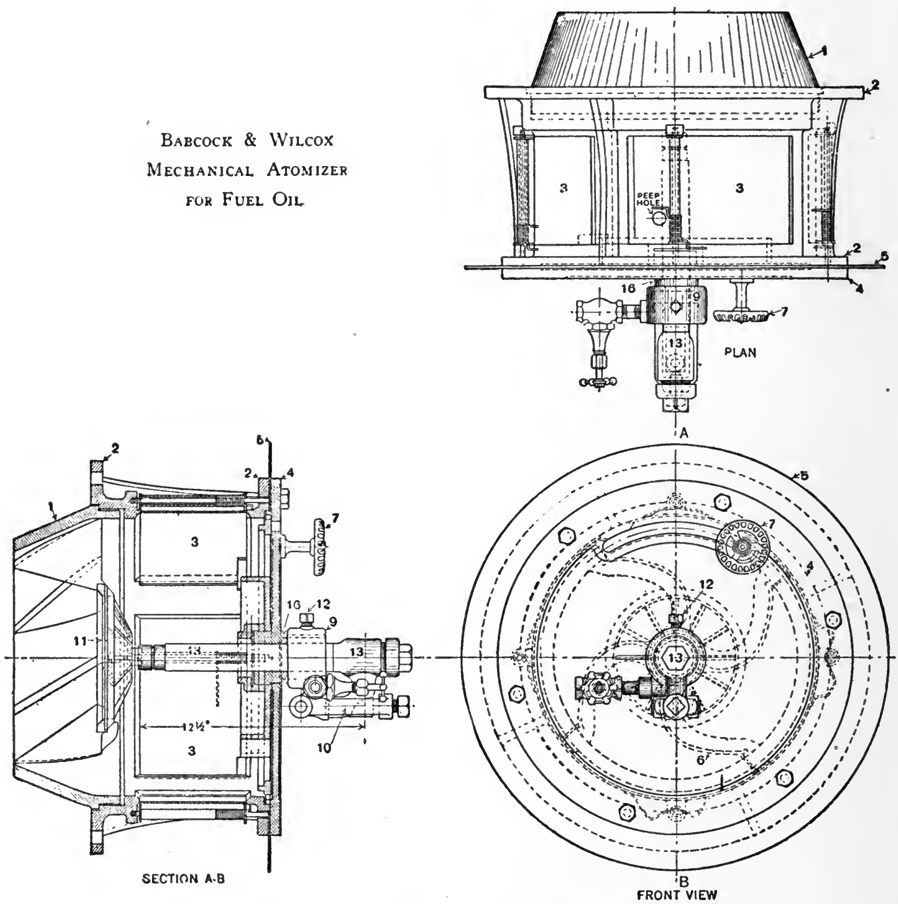



Fig. 33.—Babcock & Wilcox atomizer.

cated cone. The operator controls the furnace conditions by the adjustment of the boiler dampers, the air supply of the burners, and the pressure and temperature of the oil.

**58. Furnaces for Burning Oil.**—The furnace design is an important factor for boilers using fuel oil. Grates, furnace doors and other fittings installed when coal is used are, of course,

not fitted. The combustion chamber must be large enough so that the flames of the burning oil do not come in contact with the brickwork or boiler heating surface, otherwise carbon will be deposited on the brickwork and tubes and smoke will result. The furnace must be lined with refractory brick that will radiate the heat and assist combustion (Fig. 18). The refractory brick is backed up by insulating brick to keep the heat of the furnace from radiating into the fireroom. This brickwork is securely fastened to the boiler casing by special bricks to which hook bolts are attached. The Foster water-tube boiler has a lining of 9-in. firebrick backed by two thicknesses of 2 1/2-in. insulating brick. (See Fig. 15.) This arrangement keeps the furnace casing below 100° at the hottest part. In naval boilers in order to save weight, the thickness of the brickwork is greatly reduced and the radiation from the furnace walls is taken care of by the circulation of air in a space left for this purpose between the furnace walls and casing. With express boilers the thickness of the furnace brickwork is reduced to 4 1/2 ins. against 14 ins. used in the Foster boiler.

**59. Air Requirements with Fuel Oil.**—The air for combustion is admitted at high velocity through specially designed tuyeres or registers surrounding the burner.

The theoretical air required for combustion can be computed from the formula given in Art. 46, namely,

$$\text{Pounds of air per pound of oil} = 11.6C + 34.2(H - O/8) + 4.35S.$$

Because of the larger percentage of hydrogen present, oil theoretically requires more air than coal. The exact amount varies with the analysis of the fuel but is generally very close to 14 lbs. of air per pound of oil. This is equivalent to 184 cu. ft. of free air at 60°F. The amount of excess air necessary is much less than required for coal because of the facility with which the oil spray and air mix in the furnace. With careful management 10 per cent excess air is sufficient and 30 per cent should be the maximum for good performance. This small amount of excess air accounts in a large measure for the increased boiler efficiency with fuel oil. The reader should refer to the heat balances, Table VIII, for a comparison of the losses with oil and coal burning boilers. Naval vessels use a larger amount of excess air than necessary for good performance in order to do away with smoke.

Fuel is here sacrificed for military reasons. No air should be admitted to the boiler except at the register; air leakage in the casings should be avoided. The correct amount of air can often be ascertained by the behavior and color of the flame.

**60. Method of Operating with Oil Fuel.**—The following description of the method of operating an oil fired boiler is taken from an excellent paper by J. J. Hyland, U. S. N., in the Journal of the S. N. E., May, 1914:

> Suppose no steam is on the plant, the boilers all being cold and no steam or air available. A hand pump is connected up to the discharge end of the burner line and the hand-pump suction taken from a 5-gal. can filled with oil. The pump is started and the pressure brought up to and kept at about 200 lbs. per square inch, the pressure being kept as steady as possible. For this purpose, the three-crank pump of the Schutte-Koerting Co. is a very good one. A torch made of a hooked rod with either waste or asbestos-ball wicking on the end is dipped in the oil. The addition of a little kerosene or gasoline to the torch improves matters. The torch is lighted; all the registers are opened wide to give as much air as possible to the boiler. Fortunately oil-burning boilers are not provided with dampers, which would be a menace in unskilled hands. The burner is opened and lighted and the pump kept going constantly. Unless the oil is a light oil whose viscosity is 8 or below, at 70° F., the torch will have to be kept up to the burner constantly until the steam forms and the viscosity of the oil is reduced by heating to 8 or below, when the torch may be withdrawn. When steam pressure has reached 75 or 100 lbs., put steam on the oil pumps and heaters and cut out the hand pump. Put steam on the forced-draft blower and get ready to put on other burners; start blower slowly, open air registers wide and light burner. As soon as it is lighted, speed up the blower until you are sure the air pressure is sufficient to prevent vibration and flarebacks. Then set the vanes to the proper openings on the air registers. Always run with the air registers throttled down—it gives a better regulation and a higher velocity to the air and ensures smoother running.
>
> Until the boiler furnace is well heated an excess of air will be needed to prevent smoke. When the furnace becomes heated the air can be cut down to the proper amount. Cut in other burners as needed, first being sure the register is set and open to the proper amount. In case of vibration cut out a burner till the blower can be speeded up to give sufficient air. Run with very light smoke. Keep a half glass of water in the boiler; if the water level is too high the boiler will prime. It is a well known fact that as burners are added to a boiler, forcing it more, the water level rises, and vice versa. This should be provided for in operating the feed pumps.

Never cut out all the burners at one time on a boiler operating at full speed. The water level will drop out of sight and give you many an anxious moment until you get it in sight again. In addition, operating with oils of 21 to 17° B. gravity, the relief valve of the oil pump shows a decided tendency to stick. If you shut down a boiler suddenly with the oil pump running, there is danger of bursting the oil lines and causing a fire that might ruin the ship. Slow the pump down until the pressure is low enough to cut it out altogether; then close the burner valves.

Merchant ships generally start up when cold by using the steam generated by the donkey boiler or by a supply of steam from shore or another ship. A high pressure is needed in starting with cold and heavy oil. When steam is available it is turned onto the oil heater and the oil is recirculated through the piping and heater, the burners being closed until the viscosity has been reduced sufficiently to insure successful atomization in the burners.

When reciprocating pumps are used for fuel oil it is necessary to install a large air chamber to prevent fluctuations in pressure at the burners. Because of the necessity of a steady pressure at the burners, reciprocating pumps are not very satisfactory. The latest and best practice is to use the Quimby screw pump, or one of similar type, that gives a steady and continuous supply of oil to the burners.

# CHAPTER VI

## DRAFT

**61. General.**—In order for the air required for combustion to enter the furnace and for the products of combustion to flow over the heating surface and up the stack there must be a difference of pressure, or draft, between the air in the fireroom and the gases in the uptake. This difference in pressure is necessary to overcome the resistance offered by the fuel bed, the boiler baffling, and uptakes. Boiler draft is analogous to the flow of water in hydraulics—a difference in pressure head is required to create velocity head and overcome the lost heads due to pipe friction, valves, sudden enlargement, etc. The draft may be produced by building up a positive pressure in the fireroom or ashpit greater than that of the atmosphere, or by reducing the pressure in the uptake below that of the fireroom. The draft, or difference in pressure, must be sufficient to overcome the resistance of the fuel on the grate ($R_F$); the resistance of the boiler passages ($R_B$); the resistance through the superheat ($R_S$) and resistance through the Howden air heater ($R_H$), if one is fitted.

Draft may be either natural or mechanical. Natural draft is produced by the suction of a stack or by the wind, while mechanical draft is an artificial draft produced by fans, blowers or steam jets. Mechanical draft on shipboard is always produced by fans or blowers. Steam jets find no use whatsoever on account of the large expenditure of steam which would require large quantities of make-up feed. The railroad locomotive is a common example of draft by steam jet, where the steam from the cylinders is exhausted into the stack. Mechanical draft is divided into two types: *Forced draft* where a pressure is built up in the ashpit greater than the atmosphere, and *induced draft* where the pressure in the uptakes is reduced by a fan which draws the gases through the boiler and forces them up the stack. As the difference in pressure required for boiler draft is very small it is customary to express it in inches of

water instead of pounds per square inch. This gives a very convenient means of measuring draft.

**62. Natural Draft.**—Natural draft is similar in principle to induced mechanical draft. It is produced by the difference in weight between the column of hot gases in the stack and a column of cold air of the same height as the stack. Natural draft varies with the height of the stack, the temperature of the stack gases and the temperature of the outside air.

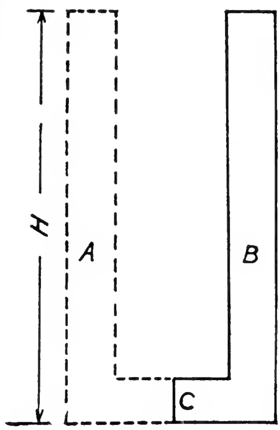


Fig. 34.

Consider an apparatus similar to that shown in Fig. 34. Let $A$ be a long pipe, $H$ ft. high, filled with cold air at temperature $t_a$ and $B$ a pipe of the same height and diameter filled with hot gases at temperature $t_s$. At $C$ a flexible diaphragm is fitted as shown. The weight of a cubic foot of air at 32°F. and at sea level is .0807 lb.; the weight of a cubic foot of stack gas under similar conditions is approximately .085 lb. The weight, or pressure exerted, by the column of air, $A$, is,

$$P_a = \frac{.0807 \times 492}{(460+t_a)} \times H \text{ pounds per sq. ft.}$$

The pressure exerted by the column of hot gases $B$, is

$$P_s = \frac{.085 \times 492}{460+t_s} \times H \text{ pounds per sq. ft.}$$

The difference in pressure on the diaphragm $C$, due to the heavier column of air, is

$$P = P_a - P_s = \frac{(.0807 \times 492)H}{460+t_a} - \frac{(.085 \times 492)H}{460+t_s}$$

$$P = H\left(\frac{39.6}{460+t_a} - \frac{41.8}{460+t_s}\right) \text{ lbs. per sq. ft.}$$

If this is converted to inches of water, we have to divide by $\frac{62.5}{12} = 5.2$ lbs. (1 inch of water pressure being equal to 5.2 lbs. per sq. ft.).

$$P = H\left(\frac{7.63}{460+t_a} - \frac{8.05}{460+t_s}\right) \text{ inches of water}$$

If the diaphragm at $C$ in Fig. 34 is replaced by a furnace and fuel bed that will maintain the stack $B$ at a temperature $t_s$ higher than the outside air we have an example of draft created by a stack. The pipe $A$ can be removed for the column of cold air equal in area and height to column $B$ will still exist.

The above formula can now be used to calculate the difference in pressure or draft measured in inches of water created by a stack.

Suppose, for example, air is at 70°F., gases leave the boiler at 500°F., and the height of the stack is 100 ft. The draft under these conditions is,

$$P=100\left(\frac{7.63}{530}-\frac{8.05}{960}\right)=0.60 \text{ in. } H_2O$$

The formula may be transposed to give the height of stack $H$ in feet necessary to produce a required draft,

$$H=P\div\left(\frac{7.63}{460+t_a}-\frac{8.05}{460+t_s}\right)$$

Thus, if 3/4 in. draft were necessary with stack gases at 550° and outside air at 60°F.,

$$H=\frac{0.75}{\frac{7.63}{520}-\frac{8.05}{1010}}=114 \text{ ft.}$$

The height given by the above formula is the theoretical height without friction. It is customary in using the formula to take the temperature of the gases at the base of the stack which is known, instead of the mean temperature in the stack which is generally unknown. In practice the height of the stack is increased about 25 per cent to allow for the friction losses in the chimney and to allow for the fact that the mean stack temperature is lower than that at the base of the stack. Likewise, for a given height of stack the actual draft obtained would be about 80 per cent of that obtained by the formula. The formula now becomes,

$$\text{Actual } P=.80H\left(\frac{7.63}{460+t_a}-\frac{8.05}{460+t_s}\right) \text{ inches of } H_2O$$

$$\text{Actual } H=1.25\ P\div\left(\frac{7.63}{460+t_a}-\frac{8.05}{460+t_s}\right)$$

The movement of the air across the top of the stack, due to the speed of the ship or a head wind, will increase the draft appreciably; a following wind may reduce the movement of air across the top of the stack to zero and hence give a draft due only to the height of the stack. Likewise, a low outside temperature ($t_a$) will increase the draft as shown by the formula. However, it is not wise to rely too much on these factors, for the full required draft should be obtained on a summer's day with a following wind.

The intensity of natural draft, therefore, is dependent solely on the height of the stack above the grate and the temperature of the gases leaving the boiler. As high boiler efficiency requires a low exit temperature for the gases, we see that a strong draft is only created at the expense of boiler efficiency. However, as was pointed out in Art. **39**, the gases leaving the boiler must always be at a temperature of at least 100° higher than that of the steam. For 250 lbs. pressure this would be about 500°F. The temperature of the gases leaving the boiler, in practice, ranges between 500° and 550°F. for boilers operating at good efficiencies.

The argument has often been advanced for stationary plants that natural or "chimney draft" is obtained only at the expense of the heat in the flue gases. Where economizers can be used for heating the feed water and the gas temperature reduced to 300° this argument holds good; and chimneys require energy to operate them as well as fans. For marine installations, however, where economizers are not practical (see Art. **182**) natural draft can be obtained without cost.

As the intensity of natural draft is fixed for good operating conditions by the height of the stack, for an ordinary cargo ship the draft is thus limited to about 1/2 in. of water. For high drafts or where any flexibility is desired, mechanical draft must be used.

**63. Forced Draft.**—Forced draft consists in creating a positive pressure greater than that of the atmosphere in the ashpit in coal burning boilers or around the registers in oil burning boilers. For coal burning boilers the draft may be produced either by the closed ashpit method or by the closed stokehold method. In the former, the ashpit is sealed tight and put under pressure by the blower; in the latter the whole stokehole is sealed and put

under pressure. The closed stokehole system is universal in naval vessels but used very little in the merchant service. This system has many advantages but has one serious disadvantage, in that the approach and exit from the fireroom can only be made through air locks.

In the closed ashpit system the pressure in the ashpit and furnace is greater than that in the fireroom; so unless the draft is shut down when opening the furnace doors the gases and flames will be blown out into the fireroom causing difficulty and serious danger in stoking. In the closed stokehole system, the whole fireroom is under the same pressure as the ashpit. Due to the loss through the fuel bed the pressure in the furnace is slightly less than that in the fireroom. Under these conditions upon opening the furnace doors there is a rush of air from the fireroom into the furnace instead of a blast of flame and gases outward. There is a small loss due to the dilution of the gases, and a larger quantity of air must be supplied because of the entrance of air through the furnace doors. With natural draft the pressure in the fireroom is also greater than that in the furnace so that there is no outward blast upon opening the furnace doors.

**64. Induced Draft.**—With induced draft the fans are placed above the boilers and draw the hot gases from the uptakes and deliver them up the stack under pressure. Induced draft is thus an intensified natural draft and has the advantages of natural draft, namely, that the draft does not have to be shut down when firing. Because of the larger volume of gases handled by the fans in the induced draft system the fans are nearly twice the size of those used with forced draft. The density of the gases, however, is less; so the fan horsepower is no greater.

**65. Types of Draft with Fuel Oil.**—When burning oil the air may be brought to the air registers from the fans through ducts or false boiler fronts; or the closed stokehole system may be employed. Natural draft and induced draft lend themselves readily to oil burning because of the simplicity, and the absence of all draft ducts and equipment in the fireroom. Induced draft was installed to a very large extent during the recent war due to the unfamiliarity of the engine force with water-tube boilers.

**66. Howden System of Forced Draft.**—In this system the air for combustion is forced by a fan over the outside surface of a

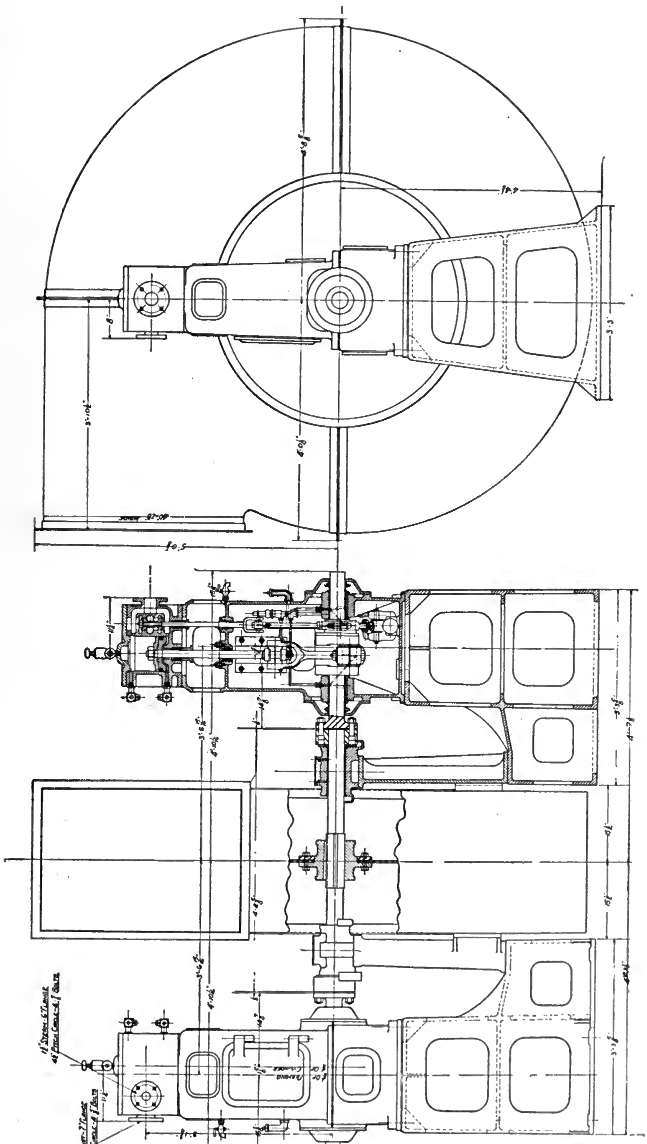


Fig. 35.—84-in. Fan, Howden forced draft.

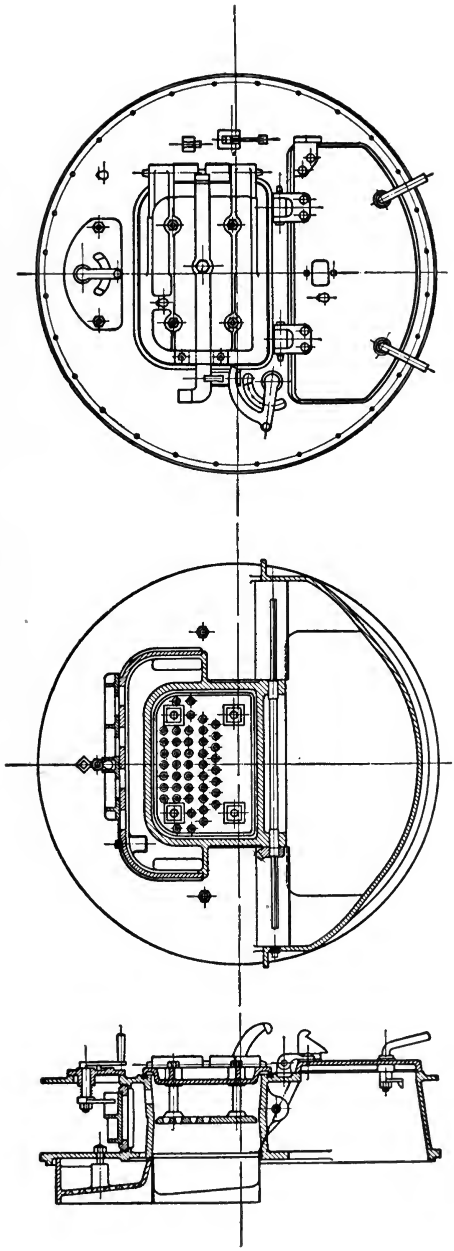
nest of tubes located in the uptake. The products of combustion after leaving the boiler pass through the inside of these tubes on the way to the funnel. This nest of tubes is known as a Howden air heater and is clearly shown in Figs. 11 and 32. The air from the fan after leaving the air heater passes down through passages on either side of the uptake into a reservoir surrounding the furnace fronts (Figs. 36a and 36b). From here the air is admitted to the ashpit and above the fire by valves in the furnace front. In principle, the system is similar to forced draft with closed ashpit already mentioned. The essential features of the system are the air heater which raises the temperature of the air for combustion to about 250°F. and the special furnace front for admitting air above and below the fire.

The absorption of the heat from the waste gases by the air for combustion, increases the efficiency and capacity of the boiler, improves the combustion, and, because of the more uniform temperature maintained in the furnace, increases the life of the boiler. The furnace fronts acting as air reservoirs prevent radiation to a large extent from the furnaces and thereby give a cooler fireroom.

The furnace front for coal burnng boilers is fitted with a safety guard so that the furnace doors cannot be opened until the air supply to the ashpit has been shut off. This prevents any danger of the flame blowing through the furnace door when it is opened for firing. When the furnace doors are open there is no air supply to the ashpit but hot air is admitted through the valves to the top of the fire. This air is passed from the valves vertically downward across the opening forming a sort of screen and thereby prevents the entrance of cold air. This arrangement is a great improvement over natural or induced draft and the closed stokehold.

The Howden forced draft system is adopted almost universally when mechanical draft is used with Scotch boilers. Altogether about 20,000,000 I.H.P. have been fitted with Howden draft. The heat in the waste gases is recovered and put to an economical use thereby reducing the fuel consumption. Howden air heaters can be used successfully even when waste type superheaters are fitted below them in the uptake, Fig. 11.

Howden draft is used with oil fuel as well as coal. Figures 32 and 36b show the furnace front and arrangement of air admission



HOWDEN PATENT FORCED DRAUGHT FURNACE FRONT
ARRANGED FOR COAL BURNING ONLY

FIG. 36*a*.

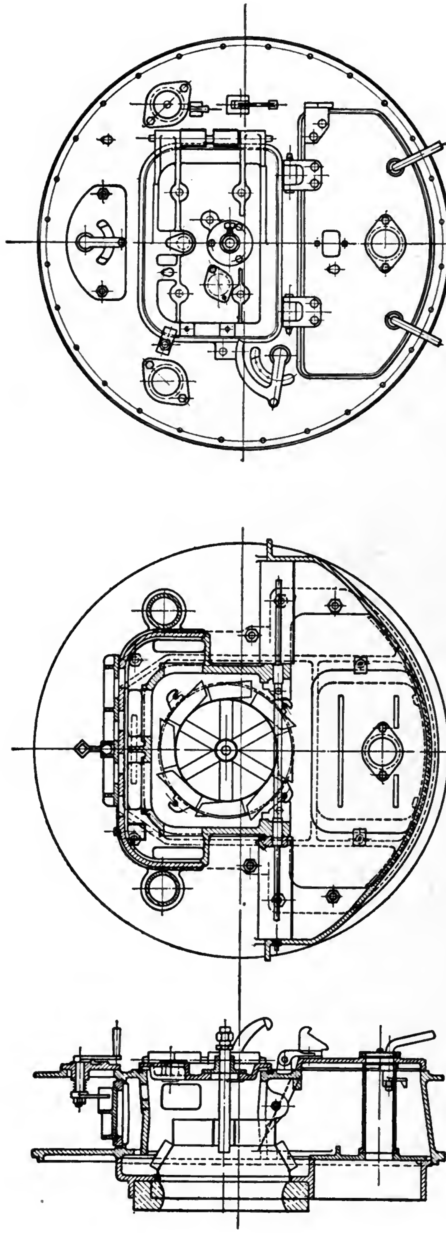


FIG. 36b—Howden forced draft furnace front arranged for fuel oil.

when using fuel oil. This system is especially advantageous with Scotch boilers using fuel oil because of the higher temperature in the uptake.

**67. Fans.**—Either slow speed plate fans are used or the higher speed type similar to the Sturtevant "Multivane,"

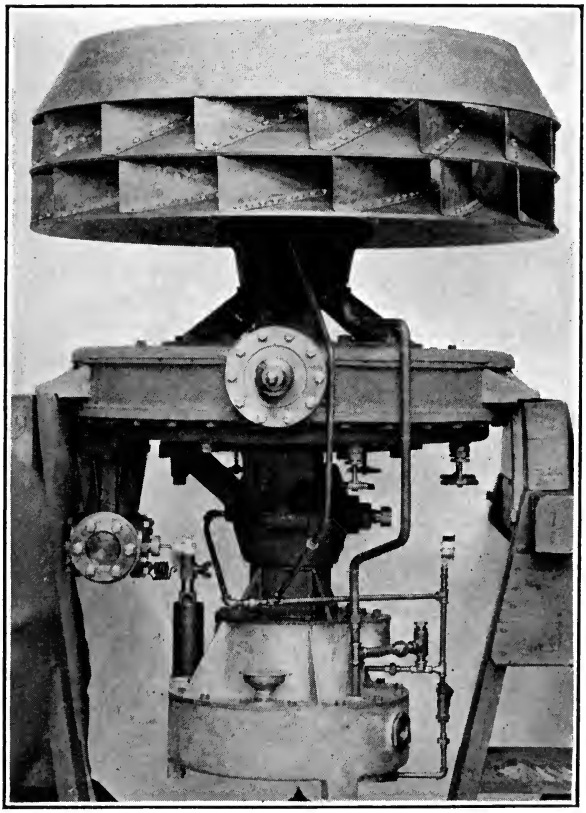
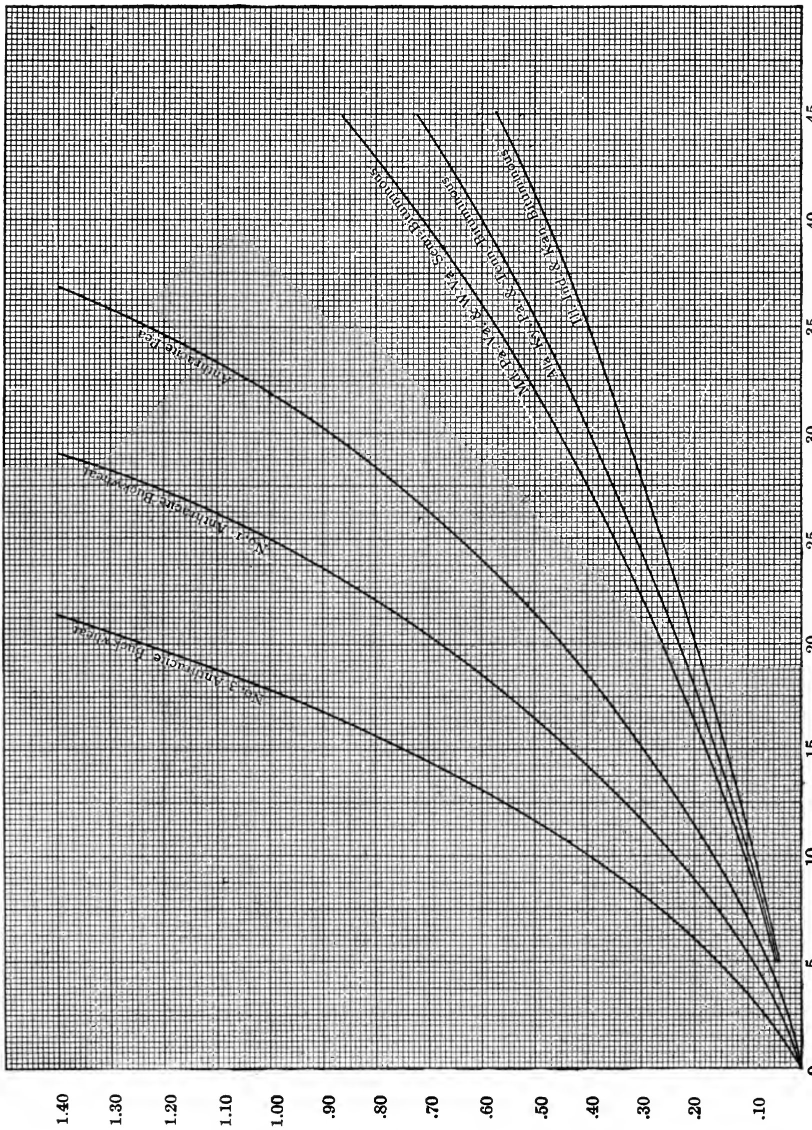


Fig. 37.—Sturtevant turbine-driven fan as used on destroyers.

depending on the space available and the amount of air to be handled. Two reciprocating engines are as a rule fitted with each fan so that one can be overhauled without shutting down the fan (Fig. 35). The fans for the forced draft as a rule are located above the boilers or high up in the engine room. By withdrawing the air supply from the engine room it helps to ventilate and cool the engine room.

Figure 37 shows a type (Sturtevant) of high speed fan used on destroyers. This fan is driven at high speed by a direct con-




Courtesy Babcock & Wilcox Co.

Fig. 38.—Draft required with various kinds of coal.

nected turbine and supplies large quantities of air under high pressure. This fan discharges air around the whole periphery

instead of at one place on the periphery as is the case with the ordinary type of fan. The fan illustrated runs at 1,350 r.p.m. and delivers air at 7 in. static pressure.

**68. Draft Required when Burning Coal.**—The intensity of the draft required varies with the rate of combustion, for as the rate of combustion is increased the fuel bed becomes thicker and a larger volume of gas is passing through the boiler. The required draft in inches is given by the equation,

$$P = R_F + R_B + R_S + R_H.$$

The resistance of the fuel bed ($R_F$) is given in Fig. 38 for various types of coal. This data has been compiled from experiments by the Babcock & Wilcox Co. The resistance or loss of draft through the boiler ($R_B$) varies with the type of boiler and the rate of combustion. The following figures cover the range of hand firing:

| | |
|---|---|
| Water-tube boilers...... | .10 − .30 in. |
| Scotch boilers.......... | .20 − .40 in. |

The loss ($R_S$) through waste type superheaters is about 0.20 in. and the loss through Howden air heater including ducts and dampers is about .30 in.

*Example.*—Find the draft required to burn 20 lbs. of coal per sq. ft. G.S. per hour in a scotch boiler fitted with a Howden heater and waste type superheater.

From curves, Fig. 38, the loss between furnace and ashpit for semi-bituminous coal = 0.26 in. .26 in.

| | |
|---|---|
| Loss through boiler = .34 | .34 |
| Loss through superheater = .20 | .20 |
| Loss through Howden air heater = .30 | .30 |
| Total draft required | 1.10 in. |

**69. Draft Required when Burning Oil.**—All the air for burning fuel oil is admitted at the registers surrounding the burners. As already pointed out it is very important that air should not be admitted at any other point. Mechanical oil burners require a fairly high velocity of air at the registers so that the air and oil spray will mix properly in the furnace. Air pressure is required to create this velocity at the tuyeres as well as overcome the losses through boiler passages, superheaters, etc. Natural

draft, forced draft and induced draft are all used with oil fuel. At low rates of combustion natural draft can be used more readily with oil than with coal. When forced draft is used without the closed stokehold system the air must be conducted from the fan to the register through ducts. The Howden draft already mentioned is ideal for this. With natural, induced draft or closed stokehold force draft, air is admitted to the register directly from the fireroom.

The amount of draft carried when burning fuel oil varies over wide ranges for the same rate of combustion, depending on the furnance volume, velocity of air used and number of burners in operation. The Navy practice is to use a larger quantity of excess air for military reasons than in the merchant service and a higher air pressure is used to prevent flarebacks from gun fire. In fixing on the required air pressure recourse should be made to test data on the boiler in question.

**70. Area of Stack.**—Various empirical formulae have been devised for the area of chimney based upon boiler horsepower and height of stack. These are far from satisfactory and are not applicable to marine boilers. Good marine practice is as follows:

1 sq. ft. stack area for each 150 lbs. of coal burned per hour.
1 sq. ft. stack area for each 200 lbs. of oil burned per hour.

In naval vessels the area of the stack is generally reduced over the above. With fuel oil the practice is to allow 1 sq. ft. area for each 300 lbs. oil burned per hour and still further reductions in area can be made without reducing the boiler capacity. The area of the stack should of course vary with the amount of excess air used. The area can be easily computed by calculating the volume of gas passing up the chimney per hour and allowing a velocity of about 20 ft. per second.

## CHAPTER VII

## COMPARISON OF COAL AND FUEL OIL FOR STEAMSHIPS

In making a comparison between coal and fuel oil for generating steam on shipboard, the problem should be considered carefully from every angle, for there are many features connected with use of fuel on shipboard that have little or no influence on shore. Some of the points to consider are:

Cost
Weight
Bunker space required
Availability and future supply
Adaptability

**71. Cost Comparison.**—In making a study of the costs of the two kinds of fuel, the comparison should be based on the amount of fuel necessary to produce equal amounts of steam under the same conditions of feed temperature and boiler pressure. In order to make this comparison certain assumptions must be made regarding calorific value of the fuel and boiler efficiency. Fair average figures for calorific values are, 14,300 b.t.u. per lb. for coal and 18,800 b.t.u. for fuel oil. The boiler efficiency when operating with coal has been assumed as 69 per cent and with fuel oil as 75 per cent. These efficiencies are for operation under the best of conditions; trials will often show higher values and generally efficiencies under working condition will be lower. However, the figures given are comparable, regardless whether or not they satisfy a particular case. The weight of oil has been taken as 7.75 lbs. per gallon (58 lbs. per cu. ft.).

With the above figures, the ratio of the heating value of a pound of oil to a pound of coal becomes:

$$\frac{18{,}800 \times .75}{14{,}300 \times .69} = 1.43$$

The following table gives the costs of oil per gallon, per ton,

and per barrel over a wide range together with the cost of coal per ton corrected to the same calorific value and boiler efficiency as fuel oil. Thus, if coal is \$6.08 per ton, it would be just as cheap to buy fuel oil at \$8.70 per ton (\$.03 a gallon), as the same amount of steam could be raised in each case for the same fuel cost. In other words, one is getting more available b.t.u.'s in a ton of oil than in a ton of coal and hence would be justified in paying a higher cost per ton for the former. Both prices are for the fuel in the bunkers.

| Oil per gallon | Oil per ton | Oil per bbl. | Equivalent cost, coal per ton, corrected to same b.t.u. boiler eff. as oil |
|---|---|---|---|
| \$0.03 | \$ 8.70 | \$1.26 | \$ 6.08 |
| .04 | 11.55 | 1.68 | 8.07 |
| .05 | 14.45 | 2.10 | 10.10 |
| .06 | 17.35 | 2.52 | 12.15 |
| .07 | 20.20 | 2.94 | 14.15 |
| .08 | 23.15 | 3.36 | 16.15 |
| .09 | 26.00 | 3.78 | 18.20 |
| .10 | 28.90 | 4.20 | 20.20 |
| .12 | 34.70 | 5.04 | 24.25 |

Walter M. McFarland has given the following rule:

When the cost of coal in dollars per ton is double that of oil in cents per gallon the fuel costs of producing steam are equal.

He assumed, no doubt, slightly different boiler efficiencies and b.t.u. values, but the above table bears out the rule.

The first cost of the oil installation with the piping, heaters, pumps, etc., is greater than that for coal. There is also a small amount of steam required—about 1 1/2 per cent of the boiler output—to operate the oil pumps and heat the fuel oil which is also unfavorable to fuel oil. On the other hand the engine room force for an oil burning ship would be cut down about 30 per cent, which is an important item in operating expenses. The life of the boiler will be longer and the maintenance costs less for fuel oil than for coal.

Oil burning boilers can be operated at a higher rate of combustion than coal burning boilers as the endurance of the fireman is not a factor; hence in many cases less heating surface will be required for oil than for coal, resulting in a cheaper and smaller boiler.

The efficiency of the coal burning boiler has been taken as 69 per cent and the oil burning boiler as 75 per cent in the above comparison. This, of course, assures continuous operation. For intermittent service, duty in port, banking fires, etc., the efficiency of the coal fired boiler would be greatly reduced while that of the oil fired boiler would be practically the same. Coal fired boilers in stationary plants show as high as 70 per cent efficiency under test conditions while the average yearly operating efficiency is between 60 and 65 per cent. Thus we can claim the elimination of stand-by losses and losses in port for the oil installation.

**72. Bunker Space.**—Fuel oil occupies about 40 cu. ft. per ton against 50 cu. ft. for coal. Thus oil requires about 80 per cent of the bunker space necessary for coal. If this is corrected by the difference in calorific value and boiler efficiency we have

$$.80\times\frac{.69}{.75}\times\frac{14{,}300}{18{,}800}=56 \text{ per cent.}$$

That is, the amount of oil necessary to produce the same amount of steam under the same conditions occupies about 56 per cent of the bunker space required for coal. In addition to this, liquid fuel can be stowed much more conveniently than coal; and the double bottom and other places not accessible for coal can be readily used for oil. How much of this saving in bunker space can be used for stowage of cargo depends to a large extent on the design of the ship. An actual layout would have to be made to get comparative figures. It is evident, however, that there will be a very large saving in space available for cargo.

**73. Weight Comparison.**—A comparison of weights of the two fuels is decidedly in favor of the fuel oil. As already shown the ratio of weight is

$$\frac{14{,}300\times .69}{18{,}800\times .75}=0.70$$

This shows that fuel oil to produce the same amount of steam

as coal weighs only 70 per cent as much as the coal—or the coal weighs 43 per cent more than the fuel oil.

The saving in fuel weight gives the oil burning ship a larger net deadweight carrying capacity. This is also accompanied by an increase in hold space as previously pointed out.

For a slow ship on a short voyage the saving in bunker space and fuel weight, and to a considerable extent fuel costs, may not be very important. However, as the length of the voyage becomes greater or speed is increased both of the factors become of more and more importance.

**74. Adaptability.**—Fuel oil is much easier and cleaner to handle on shipboard, and much easier and quicker to put in the bunkers, and results in a much cleaner ship. The difference in comfort in the fireroom is beyond comparison.

**75. Availability and Future Supply.**—From the above analysis it appears evident that with the present prices of fuel, the only one to consider is fuel oil. Even with the price of oil higher per ton than coal the factors of space, weight, reduced crew, and convenience of handling would still give oil by far the advantage.

There are, however, two other factors of greater importance than those given. The first is the possibility of fueling an oil burning ship at the various ports of call; and the second and most important of all is the question of the future oil supply of the world.

Geologists of high rank tell us that at the present rate of consumption America's supply of petroleum will be falling off rapidly in 10 or 15 years. With this rapidly decreasing supply of petroleum and the increasing demands for gasoline and fuel oil, the price of fuel oil is bound to rise steadily. Another factor that will increase the price will be the demand of fuel for use in the Diesel engines of the world's growing fleet of motorships.

On account of the much smaller fuel consumption of the Diesel engine, the motorship will be able to operate long after the oil burning steamship has ceased to be an economical ship. A study of the table given at the opening of the article would lead to the conclusion that when oil reaches a value around 8 to 10 cents a gallon the coal burning ship would be more economical than the oil burning ship, especially if the price of coal reduces due to improved labor and transportation conditions.

Steamship companies and many power plants on shore are

today ordering and installing oil burning boilers and changing over existing installations from coal to oil burning. Even apartment houses and hotels are following in this wave of oil burning and are fitting oil burning equipment.

If we are to accept the figures of the geologists, it would seem that the present tendency with regard to fuel oil burning is not altogether in the right direction. Certainly, power plants and industrial installations on shore that are not affected by the weight, and space requirements of a ship and to a considerable extent the boiler room and handling difficulties, cannot long continue to use fuel oil.

Passenger ships and naval vessels can afford to use oil on account of its cleanliness and ease of handling long after cargo ships have found it uneconomical. For the cargo carrying steamship and low powered passenger ship it would seem that the solution in the future may be a return to coal fired boilers and the adoption of the mechanical stoker.

**76. Diesel Engines.**—This chapter on fuel would not be complete without some mention of the Diesel engined motorship. The motorship is rapidly replacing the cargo steamship of 15,000 dw. and under and will probably in the near future completely supersede it. This is, of course, due to the small fuel consumption of the Diesel engine, although the machinery weights and first costs are greater for the motorship.

Allowing a fuel consumption for the Diesel engine of 0.42 lb. /S.H.P. per hour and for the oil burning steamship of 1.00 lb. /S.H.P. per hour, the fuel ratio becomes $\frac{1.00}{.42} = 2.4$. The coal burning ship (hand fired), would have a consumption of 1.43 lbs. /S.H.P., if we use the same values of b.t.u. and boiler efficiency as before.

These figures give the following results:

1. Fuel for motorship weighs about 30 per cent of that of coal burning steamship and 40 per cent of that of oil burning steamship.
2. Fuel for motorship occupies 23 per cent of space of that of coal burning steamship and 40 per cent of that of oil burning steamship.

When we stop to consider that the fuel for a coal burning steamship weighs 3.4 times and occupies 4.5 times the space that the fuel for a Diesel engined motorship does, we at once realize the tremendous saving in fuel cost and weight and bunker space for the motorship.

The fuel used by most Diesel engines is of a lighter and more expensive grade than steamship "fuel oil." Also, it is more difficult to procure and in some ports of the world it cannot be obtained. This higher cost of Diesel engine fuel offsets some of the difference in fuel consumption between the oil burning steamship and the motorship. Some types of Diesel engines have been developed to operate on the heavier and cheaper fuel oils and with a wider adoption and further development of Diesel engines this difference in fuel costs will probably disappear.

**77. Future Fuels.**—In the foregoing comparison we have considered only the existing fuels and have made no allowance for the possibility of new kinds of fuel and new methods of burning fuel. A few of these possibilities should receive attention.

The first of these is pulverized coal which today is being adopted in many land installations and bids fair to be one of the most important fuels of the future. The lower grades of coal can be used for pulverizing and the cost of pulverizing and delivery to the boiler does not appear to be excessive. The danger of explosion it is claimed has practically been eliminated; so this former objection is apparently not pertinent today. It is handled in much the same manner as liquid fuel, is very flexible and gives as high a boiler efficiency as fuel oil. Operating expenses, including crushing, drying, pulverizing, and maintenance and depreciation, etc., are claimed to be less than with mechanical stoker installations.

Whether this fuel can be used satisfactorily on shipboard is questionable. On account of the large volume occupied by pulverized coal, it is not feasible to carry it in pulverized form in the bunkers. This would necessitate, then, a crushing, drying and pulverizing plant on shipboard which would require additional space, weight, and added complications. While this fuel has attractive features for stationary plants and is bound to be widely adopted in place of hand firing and mechanical stokers, it probably will not be introduced on shipboard until more compact methods of crushing and pulverizing the coal have been devised.

Another new type of fuel that appears very attractive for use on shipboard is colloidal fuel. This is a stable liquid fuel consisting of a mixture of pulverized coal and liquid hydrocarbons, about 40 per cent coal by weight.

Colloidal fuel has been tried under boilers, has shown as good a boiler efficiency as fuel oil and can be used with the regular oil burning equipment.

An interesting article on colloidal fuel by L. W. Bates appeared in the *National Marine* for July, 1920, from which the following quotation is taken:

> One may summarize the situation regarding colloidal fuel as follows: It may be made with the smaller sizes of coals too fine to be fired on grates. Thus earth-crushed anthracite and washings from rivers may be utilized. A patented process of ash-removal permits the use of large and poor culm banks. Lump sizes of coal are not required as the wastes and dust when pulverized are equally useful. It is the most compact fuel known. Whereas oil contains about 148,000 b.t.u. per gallon, an average grade of colloidal fuel, made with bituminous coals, contains 160,000 b.t.u. and grades may be made containing over 180,000 thermal units per gallon. The liquid used in colloidal fuel may be a blend of several oils and may include the tars. In colloidal fuel 20 per cent of such tar or 30 per cent of water gas tar combined with pulverized coal may be stabilized in mineral oils. In regard to cost, colloidal fuel can compete with straight oil when the cost of oil is over 2 cents a gallon and coal at its usual prices. With oil at 10 cents and tar at 5 cents per gallon and coal wastes at $3 per ton, straight oil is 50 per cent dearer than colloidal fuel made with the above materials. Since colloidal fuel is heavier than water it will sink in it and so may be stored under water-seal and thus made fireproof. A fire consuming the fuel may be quenched with water.

It certainly seems that the possibilities of this new fuel are very great, especially to those countries that have no oil, as a liquid fuel can be produced 50 per cent oil and 50 per cent pulverized coal.

Future developments may bring new fuels and find ways of producing fuel oil such as the low temperature distillation of coal. Colloidal fuel mentioned above suggests one new and important development. Barring these developments, however, it appears that the day of the coal burning steamship has not passed.

# CHAPTER VIII

## SUPERHEATERS

The theory of superheat is treated in Chap. II and the value and improved economy with superheat is taken up in Chaps. IX and X; here we are concerned with the method of obtaining superheat and the types of superheaters used.

**78. Types of Superheaters.**—There are two types of superheaters used in marine work: (1) those installed within the boiler heating surface in the direct passages of hot gases; and (2) those

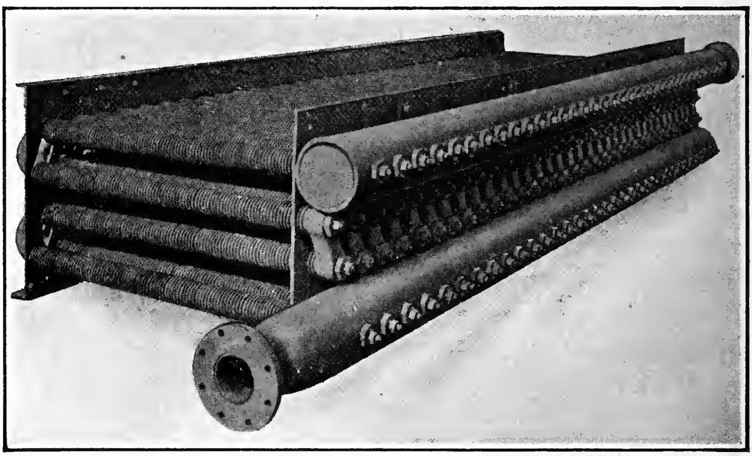


FIG. 39.—Foster waste type superheater.

installed in the uptakes and receiving their heat from the waste gases. Separately fired superheaters are used to some extent on shore but are never used in marine work. With the first mentioned type of superheater there is practically no limit to the degree of superheat that can be obtained; in marine practice, however, 250° is the upper limit used. With the waste heat type which is employed only with Scotch boilers the upper limit is about 100° while the average superheat carried is about 60°F.

**79. Waste Type Superheater.**—Figure 11 shows the arrangement of a Scotch boiler fitted with a Foster waste type superheater in the uptake. The location is such that it does not interfere with cleaning or removal of boiler tubes. This superheater is built up of seamless steel boiler tubes covered with cast iron rings. The dry or wet steam from the boiler enters one header of the superheater (Fig. 39) and after making four passes through the gas passage enters the superheated steam header and passes on to the engine room. An inner core is fitted inside the superheater tubes to insure a thin body of steam passing close to the heating surface. The cast iron rings fitted over the tubes increase the heating surface and protect the tubes in starting up and standing by.

Because of the large amount of soot collected by superheaters in the uptake, mechanical soot blowers are generally fitted with this type of superheater to keep the superheating surface clean and also to avoid loss in draft due to the choking of the passages by soot.

**80. Fire Tube Superheater.**—Figure 40 shows a Scotch boiler fitted with a fire tube superheater manufactured by The Superheater Co. This superheater consists of two headers, one for dry steam connected to the steam space of the boiler and one for superheated steam connected to the main steam line to the engine room. Steam leaves the dry steam header through a series of small U-tubes which extend into the boiler tubes. These tubes extend part way through the boiler tubes and are fitted with return bends so that one tube extends in and out of about six tubes before entering the superheated steam header. The superheating tubes are supported by small carriers inside of the boiler tubes so that they are not in contact with the heating surface. The degree of superheat obtained is fixed by the distance that these units extend into the boiler tubes. Superheat as high as 250° can be obtained with this type of superheater. Mechanical soot blowers can be fitted with this superheater; hand cleaning would be difficult and not thorough.

**81. Superheaters with Water-Tube Boilers.**—Figure 17 shows the Babcock & Wilcox type of superheater fitted in connection with the B & W boiler. This is located in the first path of the hot gases and consequently a high degree of superheat can be obtained. Figures 14 and 15 show the Foster superheater fitted

with the Foster boiler. This is also located at a place where the temperature of the gases is high and consequently a high degree of superheat is possible. The construction of the Foster super-

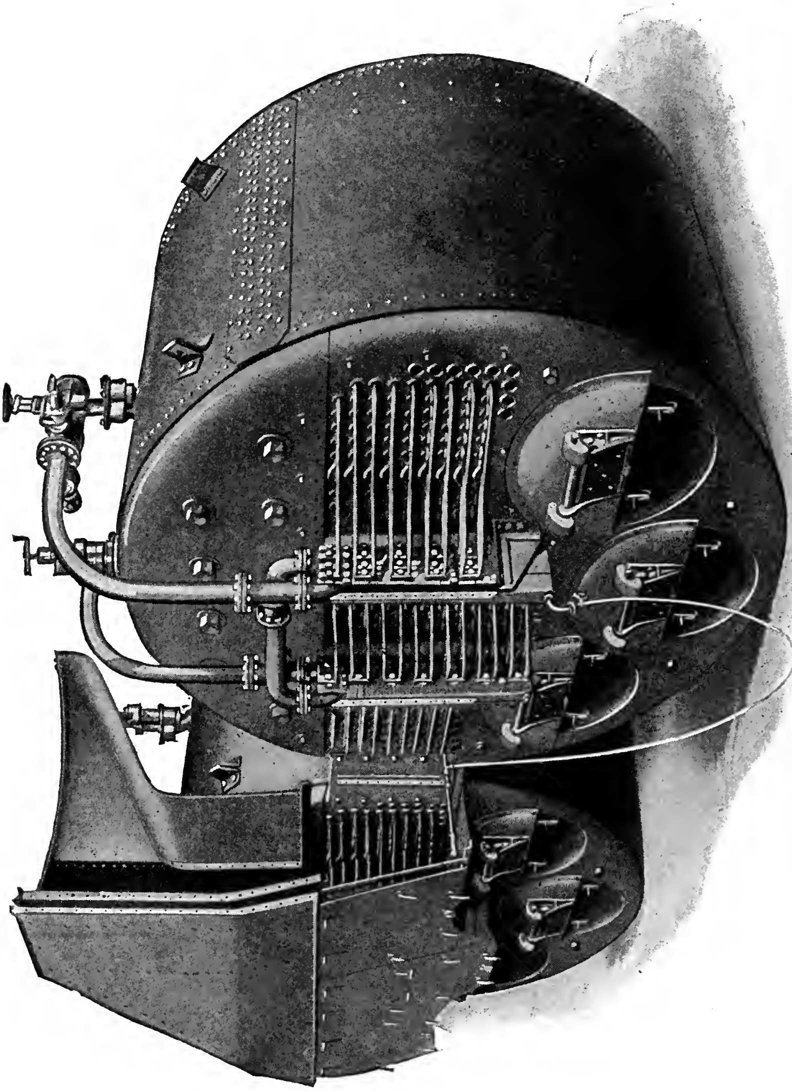


Fig. 40.—Fire-tube superheater installed in Scotch boiler.

heater is similar to that of the Foster waste heat type already described; the construction of the Babcock & Wilcox type is shown clearly in Fig. 17. So far, in marine work these types of

superheaters have been used only with 150° of superheat and no flooding arrangement has been found necessary for protecting them while starting up or standing by. Figure 21 shows a superheater fitted in connection with the Yarrow express type boiler. A damper is employed here for cutting out the passage of gases through the superheater when the main engines are not operating.

**82. Comparison of Superheaters.**—For reciprocating engine installations where only a small amount of superheat is desired, the waste type superheater carrying 60 to 80° superheat is an excellent installation. The superheat is obtained without any additional expenditure of coal and the overall efficiency of the plant is thereby increased.

With turbines where a superheater higher than 100° is advisable one of the other types of superheaters must be used. As the superheaters installed in the boiler setting absorb heat from the gases during their passage through the heating surface more fuel must be burned to obtain the superheat. The overall efficiency of the installation, however, is increased as the reduction in steam consumption more than offsets the increase in fuel consumption.

The performance of a ship using dry steam should never be compared with a ship using superheated steam on a basis of steam consumption. A true comparison can only be made on a basis of fuel per S.H.P. per hour.

**83. Superheating Surface.**—The coefficient of heat transmission for superheaters is between 2 and 5 b.t.u. per sq. ft. per degree difference per hour, depending on the velocity of the hot gases and the velocity of the steam. Allowing 3.5 as the coefficient we have

$$\text{Superheating surface} = \frac{wct_s}{3.5\,(t_1 - t_2)}\text{sq. ft.}$$

where $w$ = lbs. of steam per hour.
$c$ = specific heat of superheated steam.
$t_s$ = degree of superheat.
$t_1$ = mean temperature gases.
$t_2$ = mean temperature of steam.

*Example:* Calculate the heating surface required for a B & W type

superheater to superheat 33,000 of steam per hour to 150° superheat; pressure 250 lbs. absolute and leaving boiler as dry steam.

$$\text{S.H.S.}=\frac{33{,}000\times .58\times 150}{3.5\ (1000-475)}=1560 \text{ sq. ft.}$$

A calculation should always be made especially with waste type superheaters to see if there is sufficient heat in the gases to give the desired degree of superheat with the reduction in gas temperature anticipated. A calculation for the degree of superheat possible with a given waste type superheater installation is given in Art. **191.**

The superheater must be located so that the desired amount of superheat can be obtained. When the superheat desired is around 100° the superheater can be located at a considerable distance from the furnace as in Fig. 17. The location in the Foster boiler (Fig. 15) allows a somewhat higher degree of superheat than Fig. **17.** As higher degrees of superheat are used the superheaters will have to be located closer and closer to the furnace so that a larger temperature difference can be secured.

## CHAPTER IX

## THE RECIPROCATING ENGINE

**84. Reasons for Multi-expansion Engines.**—One of the largest losses in reciprocating steam engines is that due to cylinder condensation. As pointed out in Art. **14** this is due to the large temperature difference between the entering steam and the cylinder walls which have just been in contact with the exhaust steam. With high steam pressures and low vacua this range in temperatures becomes very great. To reduce this temperature difference in the cylinder, compounding has been resorted to. Thus, if two cylinders are used the temperature range in each cylinder is cut in half, for the steam now leaves the first cylinder at a temperature twice as high as if only one cylinder had been used. Consequently the cylinder walls are at a higher temperature when the steam enters and cylinder condensation is reduced.

Compounding has further advantages than reducing the temperature range in the cylinder. The loss due to the clearance volume is reduced as the high pressure steam now fills a much small clearance volume than before. This reduces the initial condensation and the amount of high pressure steam required to fill the clearance space. As would be expected from the foregoing, the steam consumption of the engine is reduced by compounding and greater ranges of expansion are possible. Compounding has the further advantage in that a more uniform turning moment is obtained on the propeller shaft.

**85. Triple and Quadruple Expansion Engines.**—A compound engine consists of two cylinders, a high pressure and a low pressure. As steam pressures and vacua were increased the expansion was divided up into three steps in the triple expansion engine. The cylinders in this engine are: high, intermediate, and low. And, finally, came the quadruple expansion engine with four cylinders, high, first intermediate, second intermediate, and low.

Another type of multi-expansion engine used in marine work is the four cylinder triple expansion. Here, instead of using

one large and cumbersome low pressure cylinder, the latter has been divided into two equal smaller sized cylinders. The engine of the "Delaware," Fig. 43, is of this type. Instead of

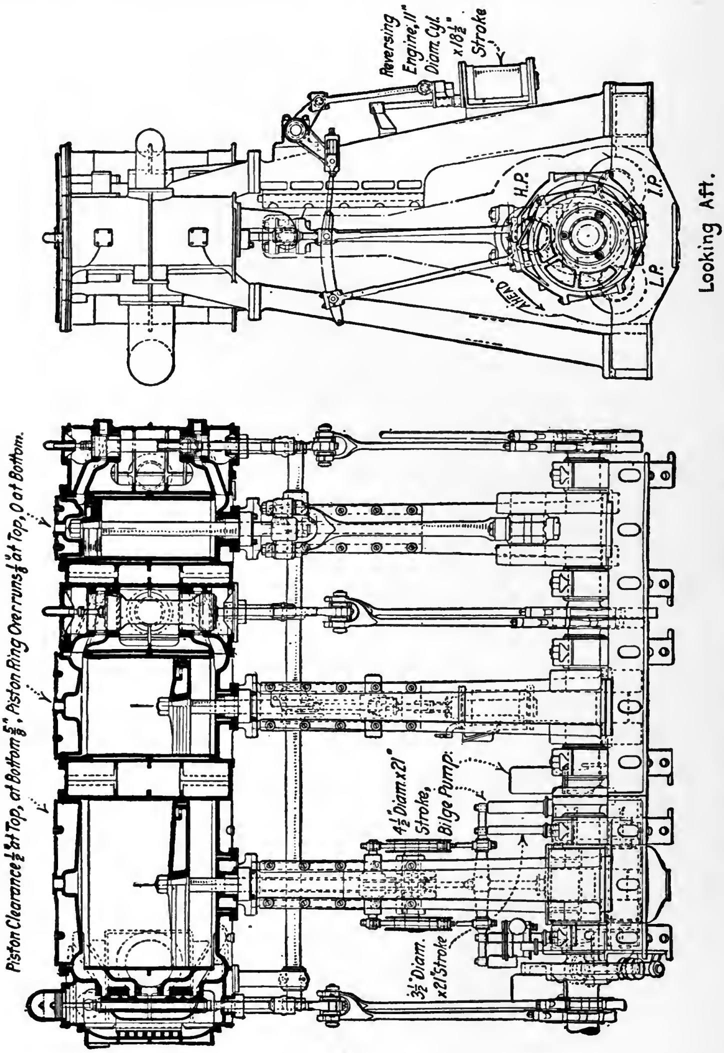



FIG. 41.—Triple expansion engine. Newport News S. B. & D. D. Co.

using one large 107 1/2-in. L.P. cylinder, two 76-in. cylinders have been substituted; the steam on leaving the intermediate cylinder divides, half flowing to the forward L.P. cylinder and half to the aft L.P. cylinder. This has the advantage of reducing the

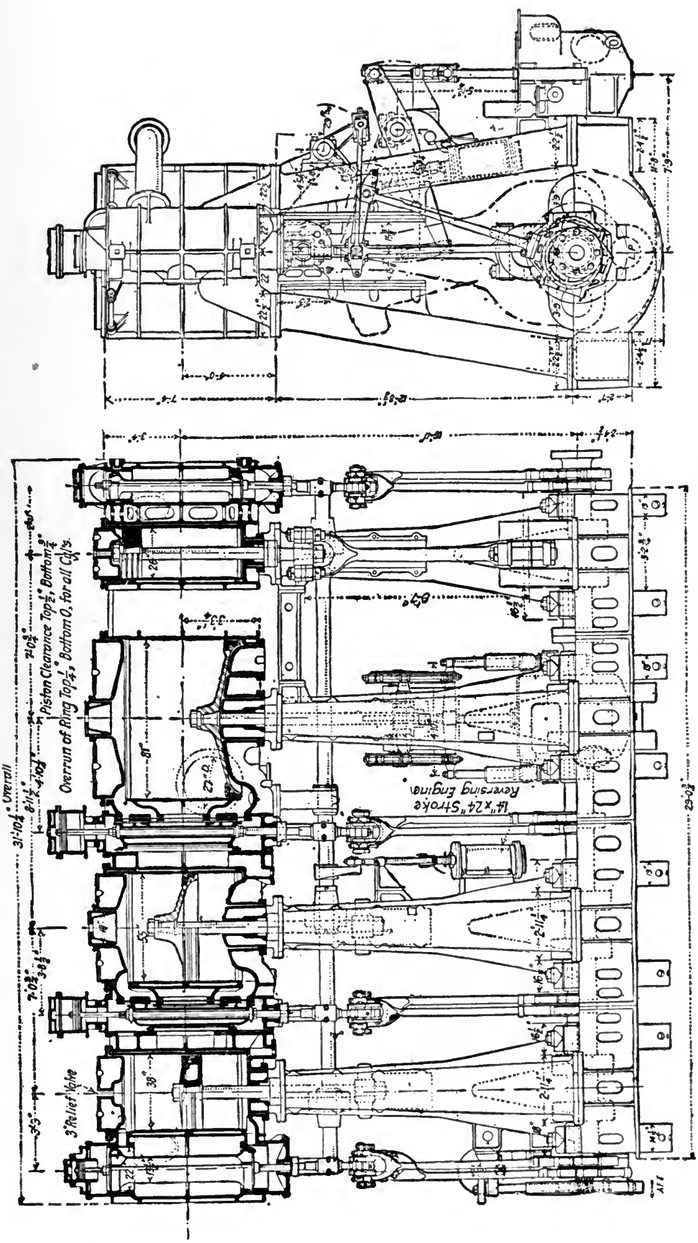
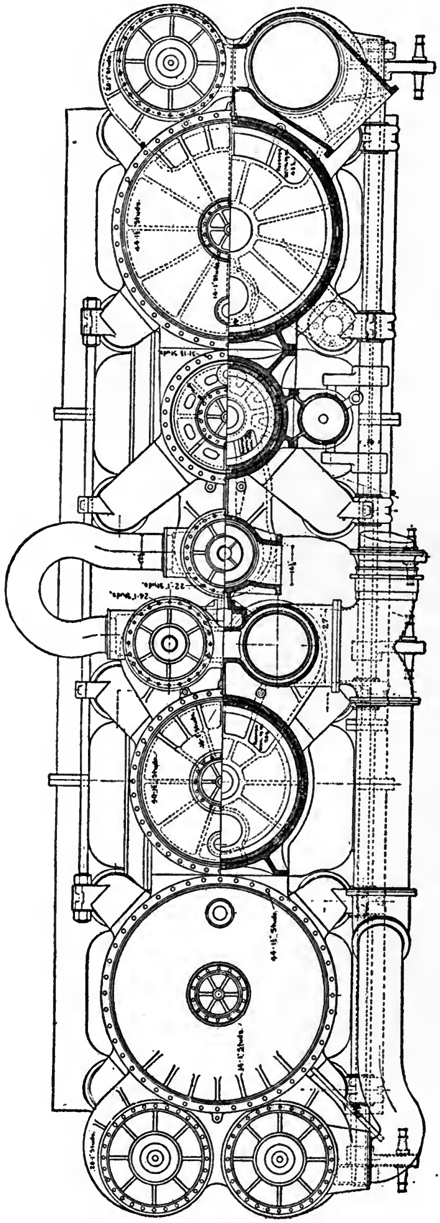



Fig. 42.—Quadruple expansion engine. Newport News S. B. & D. D. Co.



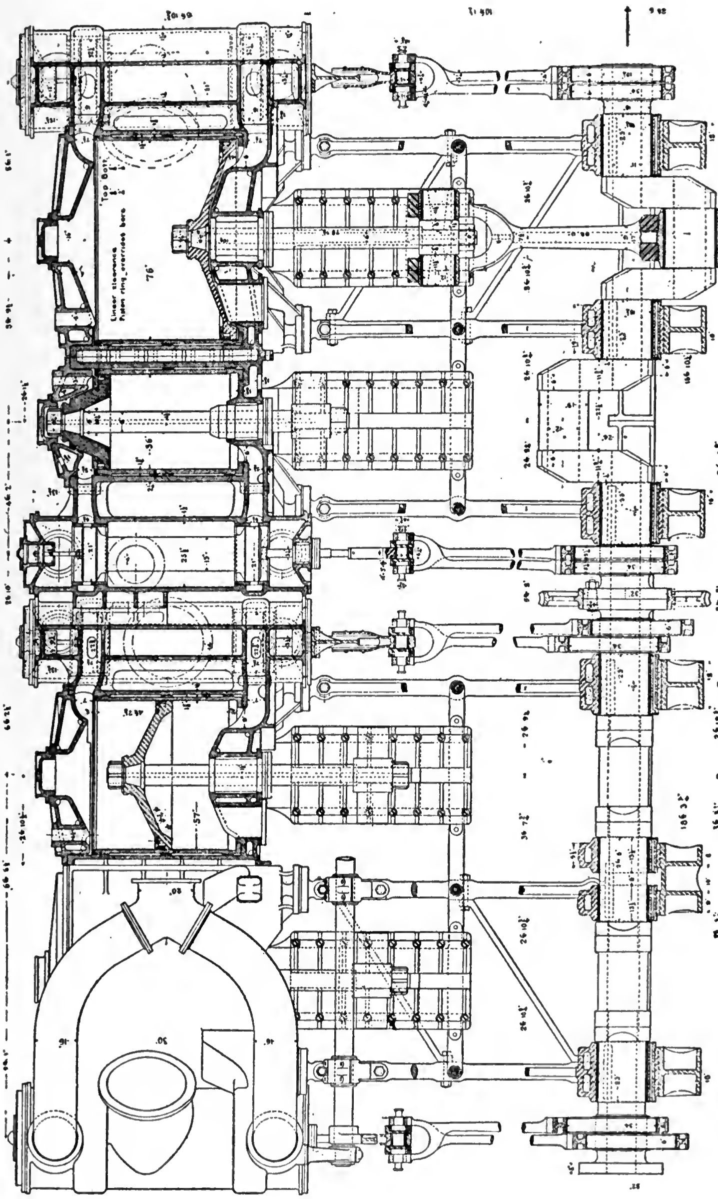


FIG. 43.—Four-cylinder triple expansion engine—U. S. S. "Delaware," from Jour. A. S. N. E., Feb., 1912.

size of the cylinder casting, piston, valve gear, etc., and also giving a more uniform turning moment.

Figure 41 shows a drawing of a triple expansion engine and Fig. 42, a quadruple expansion engine. Both of these are typical merchant engines with cast iron columns or housings of box section. The ratio of crank to connecting rod is 1:4.5. Figure 43 is a four cylinder triple expansion naval type engine with forged steel built-up framing in place of the cast iron housing. This engine is considerably lighter than the merchant type, runs at higher r.p.m. and is considerably lower. The ratio of crank to connecting rod is 1:4. The engines shown in Figs. 41 and 42 have cylinder liners in the H.P. and L.P. cylinders and that shown in Fig. 43 has liners in all the cylinders.

**86. Determination of Engine Dimensions.**—The low pressure cylinder in a multiple-expansion engine must be of sufficient size to handle all the steam used by the engine and consequently this cylinder is of the same size that would be used in a simple one cylinder engine working between the same initial and back pressures. Compounding does not reduce the size of the L.P. cylinder over that required for a simple engine.

In determining the proper dimensions for an engine the size of the L.P. cylinder is first fixed and then the sizes of the other cylinders determined, so as to divide the work up equally among the cylinders.

The piston speed is generally settled first and this should conform with good practice. For merchant engines the piston speed is between 600 and 1,000 ft. per minute; the higher the piston speed the lighter will be the engine. Naval engines have used piston speeds as high as 1,200 ft. per minute. The stroke is determined after the revolutions have been decided. The revolutions of the engine are governed by the propeller efficiency and for the slow speed merchant ship the r.p.m. used are generally between 70 and 90. The revolutions should be settled on by a careful study of the speed of the boat, wake coefficient and propeller efficiency.

Piston speed in feet per minute = r.p.m. $\times$ 2 $\times$ stroke in feet

$$P.S. = 2LN$$

$$\text{I.H.P.} = \frac{PLAN}{33{,}000} \text{ for single acting engine}$$

where $P$ is the mean effective pressure acting on piston in pounds per square inch.

$L$ the stroke in feet.

$A$ the area of the L.P. cylinder in square inches.

$N$ the revolutions per minute.

$P.S.$ the piston speed.

For a double acting engine with steam acting on both sides of the piston the I.H.P. becomes twice the above or

$$\text{I.H.P.} = \frac{P \times A \times P.S.}{33{,}000} \text{ for double acting engine.}$$

In working up the design of a multiple-expansion engine it is assumed that all the work is done in the L.P. cylinder and hence the "$P$" used in the above expressions is the m.e.p. (mean effective pressure) referred to the L.P. cylinder or as commonly written m.r.p. (mean referred pressure) or r.m.e.p. This mean referred pressure is used only in designing and is not the same as the m.e.p. in the low pressure cylinder of the actual engine. It is the m.e.p. that would result if all the expansion took place in the L.P. cylinder instead a third or a fourth of the expansion as is the actual case.

The theoretical mean referred pressure can be quickly obtained when the initial pressure ($P_1$), the back pressure ($P_B$), and the ratio of expansion ($R$) have been settled upon. The ratio of expansion is the number of times that the steam admitted per stroke to the high pressure cylinder expands in volume. In other words it is the ratio of the volume of the H.P. cylinder at cut-off to the volume of the L.P. cylinder. The proper ratio of expansion to use can only be fixed after some knowledge of engine design has been obtained by actual experience. Theoretically the higher the ratio of expansion used the higher will be the thermal efficiency of the engine but other considerations such as size, weight, and engine friction fix the ratio of expansion that it is advisable to use. In naval vessels this ratio is lower than in merchant vessels in order to cut down the weight of the engine and the size of the L.P. cylinder. (See Table IX.)

$$\text{M.r.p.} = P_1\left(\frac{1+\log_e R}{R}\right) - P_B \text{ [1]}$$

[1] See HIRSHFELD & BARNARD, "Elements of Heat Power Engineering," for the derivation of this formula.

The back pressure ($P_B$) is the pressure existing in the L.P. cylinder at exhaust and is somewhat higher than the pressure in the condenser.

The mean referred pressure found by the above formula is the theoretical value that would be obtained for an expansion between the limits of $P_1$ and $P_B$ under ideal conditions. Actually, the m.r.p. is much less than found by this expression due

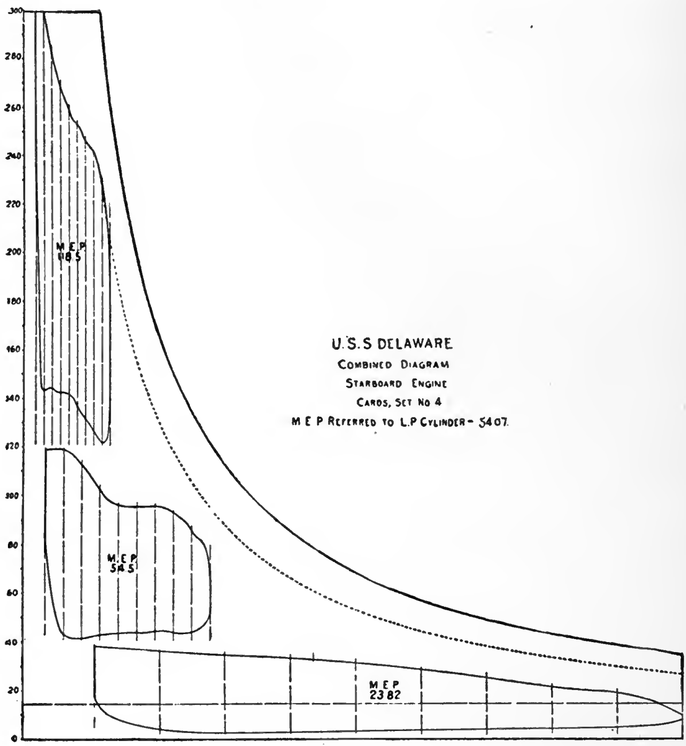



FIG. 44.

to cylinder condensation, leakage, transfer of steam from one cylinder to another, wiredrawing, incomplete expansion and clearance losses. The m.r.p. as found by above formula is multiplied by a coefficient obtained from practice. This coefficient is known as the "card effect" or "m.e.p. factor." Values obtained from engine trials are given in Table IX. Figure 44 shows the theoretical and actual cards of the U. S. S. "Delaware" from a paper by R. T. Hall in the Jour. A. S. N. E., 1909.

After the area of the L.P. cylinder has been obtained the area of the H.P. cylinder can be obtained from the ratio of expansion after the cut off of that cylinder is decided on.

$$\text{Ratio of expansion} = \frac{\text{Volume of L.P. cylinder}}{\text{Volume of H.P. cylinder} \times \text{per cent cut-off.}}$$

Corrections must be made for area of the piston rod and clearance volumes as shown in the example in the following article. The area of the intermediate piston is obtained from the L.P. and H.P. by a ratio from practice so that the work done in each cylinder is the same.

**87. Example of Engine Design.**—Determine the dimensions of a three-cylinder triple expansion engine for the following conditions:

I.H.P. = 2,900
R.p.m. = 90
Boiler pressure = 210 lbs. gage.
$P_1$ (at throttle) = 200 lbs. gage.
Superheat at throttle = 60°F.
Vacuum in condenser ($P_2$) = 27 in. (1.46 lbs./sq. in.).
Estimated steam consumption = 14.0 lbs. per I.H.P. per hour.
Back pressure in engine cylinder ($P_B$) = 3.0 lbs./sq. in.
Ratio of expansions ($R$) assumed as 11.5 (Table IX).
Piston speed (assumed) = 900 ft. per min.

$$\text{Theoretical referred m.e.p.} = P_1 \frac{(1+\log_e R)}{R} - P_B$$

$$= (215 \times .30) - 3 = 61.5 \text{ lbs./sq. in.}$$

Card effect or diagram factor = .55 (Table IX).
Actual referred m.e.p. = 61.5 × .55 = 33.8 lbs./sq. in.

$$\text{Net area L.P. cylinder} = \frac{\text{I.H.P.} \times 33{,}000}{\text{m.r.p.} \times \text{p.s.}} = \frac{2{,}900 \times 33{,}000}{33.8 \times 900} = 3{,}140 \text{ sq. ft.}$$

Diam. L.P. cylinder = $63\frac{1}{4}$ in.

$$\text{Stroke} = \frac{\text{p.s.} \times 12}{2 \times \text{r.p.m.}} = \frac{900 \times 12}{2 \times 90} = 60 \text{ in.}$$

H.P. cut-off (assumed) = .70
H.P. clearance (assumed) = 15 per cent.
L.P. clearance = 10 per cent.
Piston rod diameter (assumed) = 5 in.

$$\text{Ratio of expansion} = \frac{(2A_L - p.r)\ (1+c_3)}{(2A_H - p.r.)\ (H_C + c_1)}$$

where

$A_L$ = L.P. cylinder area.
$A_H$ = H.P. cylinder area.

$p.r.$ = piston rod area.
$c_3$ = per cent clearance L.P. cylinder.
$c_1$ = per cent clearance H.P. cylinder
$H_C$ = h.p. cylinder cut-off.

$$11.5 = \frac{(6{,}280-20)\times 1.10}{(2A_H-p.r.)\times .85}$$

$$(2A_H - p.r.) = \frac{6{,}260\times 1.10}{11.5\times .85} = 704.$$

$$2A_H = 724.$$

$$A_H = 362.$$

Diam. H.P. cylinder = 21 1/2 in.

$$\text{Ratio } (A_L/A_H) = \frac{(63.5)^2}{(21.5)^2} = 8.70.$$

$$\text{Ratio } A_L/A_I = \frac{A_L}{1.12\times\sqrt{8.70}} \quad I = \frac{3{,}160}{1.12\times 2.95} = 956 \text{ sq. ft.}$$

Diameter of intermediate cylinder = 35 in.

If desired, the ratio of L.P. area to H.P. area could have been assumed as the starting point instead of the ratio of expansions. If data from good practice is at hand either method will give practically the same dimensions.

**88. Value of Superheat.**—We have seen that a large loss with reciprocating engines was due to the entering steam giving up heat to the cylinder walls, resulting in a lowering of the quality of the steam and a consequent condensation of steam within the cylinder. If superheated steam is used, heat can be abstracted from the steam by the walls without the steam condensing. As heat passes from the steam to the relative cold walls of the cylinder some of the heat of superheat will be given up. If the amount of superheat is high enough the steam will not be lowered to the saturated condition and hence no condensation will take place. Thus the loss in steam due to condensation will be eliminated and the steam consumption of the engine decreased. Superheat also increases the initial heat contents and consequently increases the Rankine efficiency, resulting in a further lowering of the steam consumption. Superheat has the further advantage in that condensation is eliminated in the main steam line, valves and H.P. valve chest; and steam friction is reduced.

The higher the degree of superheat carried at the boiler the further it will be carried through the engine. It is possible to have so high a degree of superheat at the throttle that condensation will be eliminated in the H.P. and I.P. cylinders and also in

the L.P. cylinder up to the point of cut-off. Thus with 160 lbs. gage initial pressure and 250° of superheat an adiabatic expansion will not cross the dry steam line until it is reduced to about 15 lbs. gage. A superheat as high as this of course will greatly reduce the steam consumption and prevent loss due to condensation and leakage in all three cylinders, valves and receivers. With proper lagging radiation losses need be no higher than with dry steam. However, superheat may be carried so high that the cost of producing it plus the greater first cost and maintenance charges will offset the reduction in steam consumption. Neither steam nor fuel consumption is the true criterion for measuring the gain. The gain should be measured in reduced fuel costs minus increased capital and maintenance charges. Condensation in the L.P. cylinder and valve chest can be kept under control without superheat by using reheaters and steam jackets.

When using superheated steam, lubricating oil must be introduced into the cylinders to lubricate the cylinder walls. This requires special attention to the extraction of oil from the feed water—a problem not present when superheat is used with turbines nor when saturated steam is used. Further, high superheat requires special attention to valves and ports and passage because of the erosive effect of superheated steam. The volume occupied by a pound of steam increases as the degree of superheat is increased, consequently a larger size cylinder will have to be fitted when superheated steam is used. This may result in a slightly increased first cost of the engine.

Gray (I. N. A., 1914) advocates the use of 180 to 200° superheat with triple and quadruple expansion engines in order to carry dry steam through the L.P. valve chest. He quotes cases where a reduction in coal consumption and a consequent gain in cargo space have saved the cost of the superheater the first year. Certainly the losses in reciprocating engines in the past due to cylinder condensation and leakage around valve rods, valves and piston rods have been very large and if superheat will eliminate these losses that in itself is enough to recommend it. Gray states a case where the leakage around the I.P. and L.P. valve chest was 20 gallons per hour. Superheat as high as 200° with its additional costs, increased size of engine, and complications, hardly seems advisable at the present day, despite the gain anticipated.

The last ships in the U. S. Navy fitted with reciprocating engines carried between 50 and 80° of superheat and showed marked decrease in fuel consumption over the ships not using superheat. This amount of superheat eliminates condensation in the H.P. cylinder and yet is not high enough to cause troubles due to high temperatures. Speaking of these ships in the *Jour. American Society of Naval Engineers*, May, 1914, H. C. Dinger states:

> "The experience with superheaters on B & W boilers in the navy in connection with the latest designs of reciprocating engines has been entirely satisfactory. The superheaters have given no trouble and have required practically no repairs. No more difficulty as to internal lubrication has been experienced with superheat than there has been without it."

In view of what has been stated it seems advisable to use about 60° of superheat with reciprocating engines. With this amount an appreciable reduction in the steam consumption will result, internal lubrication will not be necessary and condensation and leakage can be eliminated in the main steam line, and H.P. chest and cylinder. This degree of superheat also allows the use of Scotch boilers with the waste type superheater. This amount of superheat will give dry steam at cut-off.

A noteworthy example of superheated steam with reciprocating engines is the Lehigh Valley R. R. Tug "Lehigh."[1] This boat is fitted with triple expansion engines and Scotch boilers. The boilers are equipped with fire tube superheaters that give between 200 and 250°F. superheat at the throttle. The horsepower of the engines was increased 27 per cent and the economy increased 17 per cent by the use of this amount of superheat. Data for this installation with and without superheat are given in Table IX. Superheat has been used on this boat and other boats of the same company for over four years and they have proved very satisfactory and practically no outlay has been necessary for maintenance. Lubricating oil is supplied to the cylinders at the rate of 1 quart in 60 hours. The temperature at the base of the stack was reduced from 527 to 385°F.—a very creditable performance with Scotch boilers.

[1] *Marine Engineering & Shipping Age*, Feb., 1922.

**89. Initial Steam Pressure.**—In Art. **13** it is shown that the theoretical steam consumption of an engine is

$$w_t = \frac{2{,}545}{H_1 - H_2}$$

where $H_1$ = the initial heat contents.

$H_2$ = the final heat contents after an adiabatic expansion.

The lower we can reduce this theoretical consumption by increasing $H_1$ the lower will be the steam consumption of the actual engine. A study of the Mollier diagram or the steam tables will show that $H_1$ can be increased both by increasing the pressure and by increasing the superheat. The Mollier diagram also clearly shows that $H_1$ can be increased more rapidly by increasing the degree of superheat than by increasing the steam pressure. Thus increasing the pressure from 275 lbs. absolute to 500 lbs. absolute (dry steam) increases $H_1$ from 1,202.8 b.t.u. to 1,210 b.t.u.—a gain of 7.2 b.t.u. The same increase in b.t.u. could be obtained by keeping the pressure at 275 lbs. abs. and adding only 9° of superheat. It will be seen that as far as initial heat contents are concerned, increase in steam pressure over wide ranges influences the initial heat contents $H_1$ very little.

There is a large gain, however, in the heat available for doing work or "heat availability" by increasing the initial pressure. Increasing the steam pressure lowers the entropy of the steam and gives a lower quality ($x_2$) at exhaust. Thus more heat is converted into useful work and less rejected in the exhaust. The work done per pound of steam $= (x_1r_1 + q_1 - q_2) - x_2r_2$. By increasing the steam pressure, $x_2$ is decreased for a given back pressure and hence the heat rejected ($x_2r_2$) is less. An example will make this clear.

| Case 1 | Case 2 |
|---|---|
| $P_1 = 500$ at 0° superheat | $P_1 = 275$ lbs. at 9° superheat |
| $P_2 = 28.83$ in. (.575 lb./sq. in.) | $P_2 = .575$ lb./sq. in. |
| $t_1 = 467.3°$ | $t_2 = 418.5°$ |
| $H_1 = 1{,}210$ | $H_1 = 1{,}210$ |
| $H_2 = 797$   $x_2 = .712$ | $H_2 = 830$ (Mollier)   $x_2 = .745$ |
| $\phi = 1.47$ | $\phi = 1.53$ |
| $H_1 - H_2 = 1{,}210 - 797 = 413$ b.t.u. | $H_1 - H_2 = 1{,}210 - 830 = 380$ b.t.u. |
| $w_t = \dfrac{2{,}545}{H_1 - H_2} = 6.17$ lbs./s.h.p. hour | $w_t = \dfrac{2{,}545}{H_1 - H_2} = 6.7$ lbs./s.h.p. |

The initial heat contents ($H_1$) is the same in each case. In Case 1 the steam pressure is 500 lbs./sq. in., while in Case 2 the pressure is 275 lbs. per sq. in. with sufficient superheat (9°) to give the same initial heat contents as Case 1. The quality ($x_2$) in Case 1 after an adiabatic expansion to 28.83 in. vacuum is .712, while, in Case 2, $x_2 = .745$. The result is that $H_2$ in Case 1 is 797 and, in Case 2, 830. The steam consumption in Case 1 is consequently 0.53 lb. (8.6 per cent) less than in Case 2.

It should also be noticed that the entropy for Case 1 is less than in Case 2, the higher "heat availability" being associated with a lower entropy. The results of the foregoing cases are very clearly brought out on Mollier and $T\phi$ diagrams. By drawing out the two cases on the $T\phi$ diagram it will be observed that the area representing the heat rejected is greater in the second case than in the first.

While superheat increases $H_1$ more rapidly than by increasing the steam pressure it also decreases "heat availability"; increase in steam pressure, however, increases both $H_1$ and "heat availability."

Pressures as high as 500 lbs. are of course not feasible today, and further we have already seen that superheat has many advantages not possible with high steam pressure. The purpose of the above discussion is to point out the gain due to high pressures that is often overlooked. For high economy both high pressures and moderate superheat are advisable.

Wire drawing or throttling is a constant heat change, the heat contents being the same after throttling as before. But due to the lower pressure the entropy is increased, the steam is superheated but the "heat availability" is reduced, resulting in less heat being converted into useful work and greater heat entering the condenser.

**90. Vacuum.**—From our expression for theoretical steam consumption it is apparent that a low vacuum will give a low value of ($H_2$) and hence reduce the steam consumption. While a vacuum of slightly over 29 in. is possible, practical considerations outweigh the theoretical gain expected. A vacuum cannot be produced without the expenditure of power and if the vacuum is carried too high the cost of producing it will offset the gain in economy and horsepower obtained.

A study of the pressures and volumes of steam at low pressures will show that the volume increases rapidly as the pressure decreases. Thus, the volume of 1 lb. of dry steam at 28 in. vacuum is practically double what it is at 26 in., and the volume at 29 in. is nearly double that at 28 in. These large

increases in steam volume can be easily taken care of in a turbine; but with a reciprocating engine they call for a large increase in the size of the L.P. cylinder and valves. This not only increases the size and first cost of the engine but also increases the area of the cylinder walls and offers greater opportunity for condensation. Further, the increased size of the cylinder, piston, and valves greatly increases the friction of the mechanism. The vacuum can be carried so high that the increase in friction in the L.P. cylinder is greater than the gain in power due to the higher vacuum.

The above facts, together with the increased power and size required for the air and circulating pumps, and greater danger of air leakage show at once that a high vacuum is not desirable with a reciprocating engine.

High vacua are often advocated for reciprocating engines and trials are quoted to show the reduction in steam consumption. First costs and cost of producing the higher vacuum are seldom mentioned however. Probably 27 in. or possible 27 1/2 in. as a maximum can be carried to advantage in the condenser. This will give 25 1/2 to 26 1/2 in. in the L.P. cylinder exhaust ports. Even with these vacua, careful attention must be given to the L.P. valve and steam passages to be sure that they are of sufficient size to take care of the large volumes of steam.

Too little attention has been given to vacuum for merchant ships and vacua around 21 or 22 in. are altogether too common. If proper attention is given to air pump capacity, air leakage, valve and steam port design, a considerable gain in economy and horsepower could be obtained by carrying a higher vacuum.

Operating engineers at times, in ignorance of the gain due to vacuum, decrease the vacuum in order to increase the temperature in the feed tank. Proper attention to the design of feed heating system would obviate this bad practice.

**91. Methods of Increasing Economy.**—In the engines for the last U. S. battleships using reciprocating engines this type of prime mover reached its highest state of development. Some of the means adopted to improve the economy of these engines are given below.

1. Forced lubrication.
2. Increased vacuum (27 in.) over merchant practice.

TABLE IX—PERFORMANCE OF RECIPROCATING ENGINES

| Ship or location | Type | Reference | Cylinder dimensions | (1 engine) horsepower and r.p.m. | Press. (abs.) at engine | Superheat | Vacuum, inches | Ratio of expansion | r.m.e.p. | Steam/i.h.p. hour. Main engines | m.e.p. factors | Efficiency ratio | Coal per i.h.p. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Land—pumping......... | quadruple[3] | *Eng. News* May 4, 1899 | $\frac{19.5\times29\times49.5\times57.5}{42}$ | 712 at 36.5 | 200 | 0 | 28 | ..... | 35.5 | 12.26 | ..... | .742 | 1.12 |
| Land—pumping......... | triple[1] | Peabody's Thermo. | $\frac{13.7\times34\frac{3}{8}\times39}{72}$ | 575 at 50.6 | 190.4 | 0 | 27¼ | 10 | ..... | 11.10 | ..... | .74 | 1.15 |
| Land—pumping......... | triple | *Eng. News.* Aug. 23, 1900 | $\frac{30\times56\times87}{66}$ | 801 at 17.2 | 185 | 0 | 28.2 | ..... | 23.4 | 10.33 | ..... | .705 | 1.09 |
| Land—pumping......... | 4 cyl. triple | *Eng. News* Oct. 2, 1902 | $\frac{32\times47\times58}{59}$ | 2,860 at 85 | 173 | 230 | 27.9 | ..... | ..... | 8.97 | ..... | .735 | ..... |
| Land—high speed....... | triple | N.E.C.I.E.&S. Morrison 1910 | $\frac{18.5\times27\times40}{20}$ | 1,080 at 250 | 202.7 | 91 | 27.9 | 11.7 | ..... | 12.7 | ..... | ..... | ..... |
| U.S.S. Delaware[2]........ | 4 cyl. triple | S.N.E. 1912 | $\frac{38.5\times57\times76\times76}{48}$ | 14,290 at 128.5 | 285 | 61.6 | 26.3 | 8.71 | 50.9 | 13.38 | .545 | .56 | 1.89 |
| U.S.S. Birmingham...... | 4 cyl. triple | S.N.E. 1912 | .................. | 7,738 at—191 | 240 | Sat. | 26.5 | 11.5 | 36.3 | 17.4 | .546 | ..... | 2.28 |
| U.S.S. South Carolina... | 4 cyl. triple | S.N.E. 1912 | $\frac{32\times57\times72\times72}{48}$ | 8,825 at 121 | 256 | 47 | 26.24 | 12.1 | 37.0 | ...... | .526 | ..... | 1.40 |
| U.S.S. Michigan........ | 4 cyl. triple | S.N.E. 1912 | .................. | 8,000 at 119 | 260 | 85.6 | 26.95 | 12.6 | 34.2 | ...... | .493 | ..... | 1.51 |
| U.S.S. Michigan........ | 4 cyl. triple | S.N.E. 1912 | .................. | 1,745 at ... | 88 | 51 | 27.35 | 12.3 | 12.35 | ...... | 527 | ..... | ..... |
| U.S.S. Cyclops.......... | 4 cyl. triple | S.N.E. 1912 | .................. | 3,368 at ... | 192 | 0 | 27.0 | 12.04 | 33.75 | ...... | .634 | ..... | ..... |
| U.S.S. New York....... | 4 cyl. triple | S.N.E. 1915 | $\frac{39\times63\times83\times83}{48}$ | 15,190 at 128.5 | 280 | 3° | 26.1 | 10.9 | 45.0 | 14.92 | .433 | ..... | 1.57 |
| U.S.S. New York....... | 4 cyl. triple | S.N.E. 1915 | $\frac{39\times63\times83\times83}{48}$ | 9,311 at 110.5 | 225 | 8° | 28.1 | 10.9 | 32.3 | 13.54 | ..... | ..... | 1.75 |
| S.S. Rampo........... | quadruple | S.N.E. 1919 | $\frac{24\times35\times51\times75}{51}$ | 2,990 at 74.5 | 232 | 0 | 27.9 | 14.55 | 34.7 | 12.8 | .582 | .597 | 1.11[4] |
| S.S. Lehigh (tug)........ | triple | *Mar. Eng.* Feb. 1922 | $\frac{16\frac{9}{16}\times25\frac{1}{4}\times43\frac{1}{4}}{30}$ | 633 at 87.7 | 188 | 0 | 23.5 | ..... | 32.3 | ...... | ..... | ..... | 2.29 |
| S.S. Lehigh (tug)........ | triple | *Mar. Eng.* Feb. 1922 | | 803 at 92.3 | 188 | 200–250° | 23.4 | ..... | 38.6 | ...... | ..... | ..... | 1.90 |

[1] Steam jackets and reheaters. [2] 12.5 lbs./i.h.p. at ⅔ power. [3] Replaced because of high maintenance cost. [4] Oil fuel.

3. Moderate degree of superheat (60°).
4. Large and straight H. P. valve passages by the use of piston valves (resulting in reduction in steam velocities and hence less drop in pressure).
5. Reduction in cylinder clearances.
6. Steam jackets introduced on L.P. cylinder walls and ends (to reduce condensation).
7. Double piston valves on L.P. cylinder.
8. Steam seal on L.P. piston rod (eliminating air leakage, water leakage and entrance of oil into cylinder).
9. Four cylinder triple in place of quadruple expansion engine with its large L.P. cylinder.
10. Increased size of L.P. valve ports.
11. Reheaters fitted (to reduce condensation).

Assisting steam cylinders are often fitted on the intermediate and low pressure valve rods to reduce the friction load on the valve gear due to the heavy valves. Figure 42 shows an engine fitted with assisting cylinders.

For details of reciprocating engines, engine design and valve gears, the reader is referred to the following books and papers:

Sterling, "Marine Engineers' Handbook."

Peabody, "Computations for Marine Engineers."

Bragg, "Design of Marine Engines and Auxiliaries."

E. H. Janson, "The Naval Reciprocating Steam Engine, Its Characteristics, Dimensions and Economics." *Jour. A. S. N. E.*, Feb., 1912.

# CHAPTER X

## THE STEAM TURBINE

The elementary principle of the steam turbine consists in converting the potential heat energy of the steam into kinetic energy by expanding it through a nozzle. The steam leaving the nozzle at high velocity is directed against the buckets of a wheel which rotates due to the impinging action of the jet and the kinetic energy of the steam is converted into mechanical work.

**92. The Flow of Steam.**—When steam under pressure is at rest its energy is potential and is in the form of heat. Thus, the potential energy of a pound of steam confined in a vessel under a pressure of $P_1$ lbs. per sq. in. is its total heat contents,

$$\lambda_1 = r_1 + q_1$$

This potential heat energy is measured above the established datum of 32°F. If the vessel containing this steam is opened up to the atmosphere or another vessel at lower pressure, flow will take place from the higher to the lower pressure and part of the potential energy in the form of heat contents will be converted into kinetic energy or velocity.

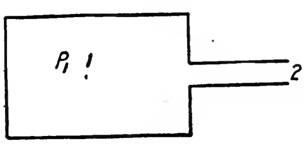


Fig. 45.

Consider the vessel in Fig. 45. Neglecting friction, we can say that the sum of the potential and kinetic energies at (1) are equal to the sum of the potential energies at (2). The total energy at (1) must equal the total energy at (2), provided there is no loss, by the law of conservation of energy. We can now set up an energy equation between points (1) and (2),

$$778\ WH_1 + \frac{WV_1^2}{2g} = 778\ WH_2 + \frac{WV_2^2}{2g}$$

where $H_1$ and $H_2$ are the heat contents at points (1) and (2) in b.t.u. per lb., $\frac{WV_1^2}{2g}$ and $\frac{WV_2^2}{2g}$ are the kinetic energy at points (1) and (2) in ft. lbs. and $W$ is the weight of steam present in the vessel. By multiplying the heat contents by 778 all the terms are reduced to a common basis of foot-pounds.[1]

Students of hydraulics will observe that by cancelling out the $W$'s, the above equation is similar in form to the Bernouilli equation with the term for gravity head omitted. In practically all cases involving the flow of steam, the difference in elevation of two points can be neglected.

If we consider that the pressure at (2) is lower than at (1), flow will take place from (1) to (2). In nearly all cases we can consider that the steam in the initial condition at point (1) is at rest; hence $V_1=0$. Cancelling the $W$ common to all terms and placing $V_1=0$ we have

$$\frac{V_2^2}{2g} = 778(H_1 - H_2)$$

The velocity of the steam at point (2) in feet per second is,

$$V_2 = 223.8\sqrt{H_1 - H_2}$$

where $H_1$ is the initial heat contents at pressure $P_1$ and $H_2$ is the final heat contents after expansion from $P_1$ to $P_2$.

**93. Expansion of Steam through Nozzles.**—In steam turbines the steam is expanded from pressure $P_1$ to pressure $P_2$ in a nozzle, Fig. 46. The expansion is rapid and there is practically no transfer of heat between the sides of the nozzle and the steam. Hence, as this is an expansion without heat being received or rejected, the velocity must be created (i.e., work done) by the heat in the steam, and the expansion must be adiabatic. This expansion is an irreversible adiabatic and is not necessarily isentropic.

FIG. 46.—Steam nozzle.

In Art. 10 one form of adiabatic expansion was discussed, i.e., throttling of steam, in which entropy increased; in fact the

[1] The term 778 $WH_1$ is in ft. lbs. for $H_1$ is expressed in b.t.u. per lb. and $W$ is in pounds; hence $WH_1$ must be in b.t.u. and 778 $WH_1$ in ft. lbs.

expansion line of the Mollier diagram crossed the lines of constant entropy at right angles. This expansion at constant heat contents (work = 0) is represented on the $T\phi$ diagram, Fig. 47, by line 1–4. Line 1–2 represents the isentropic adiabatic expansion in the cylinder of a steam engine where work done $= E_1 - E_2$.

Experiments show that the adiabatic expansions of steam in a nozzle is along a line 1–3 and very nearly isentropic. Due to friction of the steam on the sides of the nozzle the heat contents and the quality of the steam at the end of the expansion are slightly higher than after a expansion (1–2) at constant entropy.

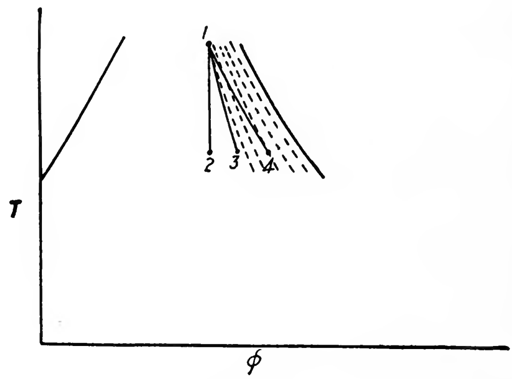



Fig. 47.

In all problems involving the flow of steam, the expansion are assumed to take place at constant entropy and a correction factor is used to correct for the final heat contents and quality. The factor $y$ is used to represent the percentage of the available heat $(H_1 - H_2)$ that is used in overcoming friction and hence represents the percentage of loss. The factor $(1 - y)$, therefore, represents the percentage of the available heat that is converted into kinetic energy. Hence, the *actual velocity* of steam issuing from a nozzle is,

$$V = 223.8\sqrt{(H_1 - H_2)(1 - y)}$$

The factor $y$ is fairly well determined from experiments and ranges between .05 and .10.

The percentage $y$ of the available heat is not lost by radiation through the walls of the nozzle but is used in overcoming friction

and is returned to the steam in the form of heat. It appears in the steam at point 3 in the form of increased heat contents. We thus see that the frictional resistance offered by a nozzle decreases the velocity of the steam and increases the final heat contents ($H_2$), quality, and entropy of the steam at the end of the expansion.

**94. Ideal or Theoretical (Rankine) Efficiency.**—In the ideal nozzle there would be no friction, the expansion would be isentropic and the velocity would be,

$$V = 223.8\sqrt{H_1 - H_2} \text{ ft. per second.}$$

Thus $H_1 - H_2$ b.t.u. per lb. of steam would be converted into useful work, the same as in the theoretical steam engine.

The expressions developed for the theoretical Rankine cycle of the steam engine (Arts. **11** to **17**) apply to the steam turbine also.

$$\text{Theoretical efficiency} = \frac{H_1 - H_2}{H_1 - q_2}$$

$$\text{Steam consumption of ideal turbine } W_t = \frac{2{,}545}{H_1 - H_2}$$

$$\text{Thermal efficiency of actual turbine} = \frac{2{,}545}{W_a\,(H_1 - q_2)}$$

$$\text{Efficiency Ratio} = \frac{W_t}{W_a}$$

**95. Thermodynamics of the Steam Turbine.**—Figure 48 shows the $T\phi$ diagram of the actual turbine. Area *1–2–3–4–5* represents the heat received ($H_1 - q_2$); area *5–6–7–4* represents the heat rejected ($x_2 r_2$), where $x_2$ represents the quality at point 6 instead of at point *8* as in the ideal case. The heat rejected in the theoretical cycle is area *8–3–4–5*. As more heat is rejected in the actual cycle than in the theoretical, less heat is converted into work. The heat converted into work is not represented by any area on the diagram but is equal to area (*1–2–3–4–5*) − area (*5–6–7–4*). This is equivalent to area (*1–2–8–5*) − area (*8–6–7–3*).

In the foregoing discussion, the turbine has been considered as a simple case of a single nozzle and one revolving wheel. The difference between the energy at entrance and exit of the

nozzle, $(H_1-H_2)(1-y)$, has been converted into useful mechanical work on the turbine shaft. Actual turbines consist of a combination of many nozzles, revolving wheels and stages, but the principles and theoretical equations given above will hold for

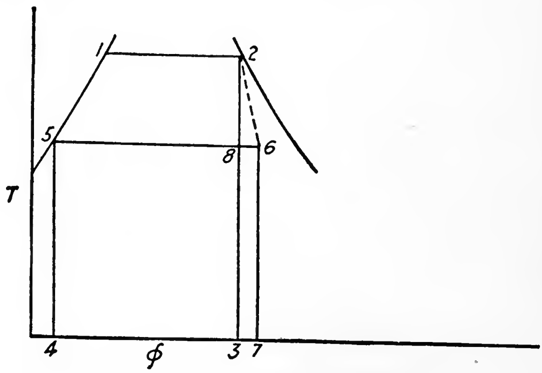



FIG. 48.—$T\phi$ Diagram of the steam turbine.

the compound turbine as well as the simple turbine. The point 1 always represents the condition at the turbine throttle and 2 the condition in the condenser.

*Example.*—Calculate the theoretical and actual velocity of steam expanded in a nozzle from dry steam at 200 lbs. abs. to atmospheric pressure, friction factor $(y) = .08$.

$$\text{Theoretical velocity} = 223.8\sqrt{H_1-H_2}$$

$$\text{Actual velocity} = 223.8\sqrt{(H_1-H_2)\times .92}$$

From the Mollier diagram $H_1$ for dry steam at 200 lbs. $= 1,198$ b.t.u.
$H_2$ after adiabatic expansion ($\phi = 1.545$) at 14.7 lbs. $= 1,010$ b.t.u.
$H_1 - H_2 = 188$ b.t.u.

$V_t = 224\times\sqrt{188} = 3,070$ ft. per sec.
$V_a = 224\times\sqrt{173} = 2,945$ ft. per sec.
$x_2 = .855$ (see $T\phi$ diagram, Fig. 47).

The heat contents $(H_2)$ after an actual expansion in a nozzle with $y = .08$ is,

$$H_2 = 1,198 - 173 = 1,025$$

where

$$173 = (H_1-H_2)\ (1-y), \text{ the heat converted into work.}$$

Therefore, $x_3r_2 + q_2 = 1,025$
where $x_3$ is the actual quality of the steam.

Solving for $x_3$ we have,

$$x_3 = \frac{1{,}025 - q_2}{r_2} = \frac{1{,}025 - 180}{970.3}$$

$$x_3 = .871 \text{ (see } T\phi \text{ diagram, Fig. 47).}$$

The quality $x_3$ can also be obtained from the Mollier diagram at $P_3 = 14.7$, and heat contents $= 1{,}025$.

$x_2$ after a pure adiabatic expansion, is .855

It will be observed that in obtaining actual velocity, theoretical steam consumption, etc., the value of $x_3$ is not required; $H_2$ is obtained as if the expansion were adiabatic, and $(H_1 - H_2)$ is corrected by $(1 - y)$ as shown in the above example.

**96. Steam Nozzles.**—If steam is to be expanded adiabatically from a high to a low pressure without irregular action, loss and dissipation of energy, a nozzle must be used in which the areas of the cross sections are made proportional to the changes in the volume of the steam during its reduction in pressure. As the heat contents of the steam is converted into velocity along the nozzle, the pressure of the steam drops and the volume and velocity increase. Thus we have three variables to determine in fixing the areas of the sections of the nozzle.

From the fundamental equation of hydraulics, $Q = av$, we have for the area of the nozzle in square inches,

$$a = \frac{Q}{V} = \frac{144\ w.s.x.}{V}$$

where $s$ = the volume of one lb. of steam in cu. ft.
$x$ = the quality.
$V$ = the velocity of the steam in feet per sec.
$w$ = the number of lbs. of steam flowing per sec.

Both analytical and experimental investigations show that the area of the nozzles reaches a minimum at a pressure equal to .58 of the initial pressure and then increases gradually again to the point of exit. Thus the area of the cross section will be a minimum for $P_2 = .58P_1$. The nozzle will first converge and then diverge as shown by Fig. 46.

A correctly designed nozzle will have a curved contour from entrance to exit. This curve from throat to exit is so nearly a straight line that it is the universal practice, for the sake of simplicity in construction, to give the nozzle a straight taper

from throat to exit. The area at entrance is indeterminate. Figure 46 shows a typical turbine nozzle. It is given a rounding curve from entrance to throat, a straight taper from throat to exit, and a cylindrical guide section between the exit and the wheel. The length is obtained by using a taper of 1 : 5 to 1 : 12 from throat to exit and a small, additional length from throat to entrance. The sections of the nozzle in practice are made both square and circular.

**97. Nozzle Design.**—The method of designing a nozzle is illustrated by the following example:

Determine the areas at throat and exit for a nozzle to expand 1,800 lbs. of steam per hour from 200 lbs. abs. to 2 lbs. abs.

$y = .10$.
$P_1 = 200$ lbs. (entrance).
$P_2 = .58P_1 = 116$ lbs. (throat).
$P_3 = 2.0$ lbs. (exit).
$W = 1{,}800 \div 3{,}600 = 0.5$ lb. per sec.
$H_1 = 1{,}198$.
$H_2 = 1{,}147$ (Mollier diagram). $\phi = 1.545$.
$H_3 =$ 898 (Mollier diagram).
$x_1 =$ 1.00.
$x_2 =$ .956.
$x_3 =$ .787.
$V_1 = 0$.

$$V_2 = 224\sqrt{1{,}198 = 1{,}147} = 1{,}600 \text{ ft. per sec.}$$

$$V_3 = 224\sqrt{(1{,}198 - 898)\ .90} = 3{,}680 \text{ ft. per sec.}$$

$$x_4 \text{ actual quality at } 3 = \frac{\mathrm{H}_1 - \{(H_1 - H_3)(1 - y)\} - q_3}{r_3}$$

$$= \frac{1{,}198 - 270 - 94}{1{,}021.9} = .815$$

From the Mollier diagram at $P_3 = 2$ lbs. and heat contents $= (1{,}198 - 270) = 928$,

$x_4 = .815$, which is more readily obtained than by the above method.
$x_1s_1 = 1.00 \times 2.288 = 2.29$ cu. ft.
$x_2s_2 = .956 \times 3.844 = 3.68$ cu. ft.
$x_4s_3 = .815 \times 173.1 = 141.00$ cu. ft.

$$a_2 = \frac{144 \times 3.68 \times .5}{1{,}600} = \frac{(144\ x.s.w.)}{v} = .166 \text{ sq. in.}$$

$$a_3 = \frac{144 \times 141 \times .5}{3{,}680} = 2.76 \text{ sq. in.}$$

**98. The Dynamics of the Steam Turbine.**—Figure 49 shows the velocity diagram for a simple impulse turbine, with the steam entering horizontally.

$V_1$ = absolute velocity of the steam leaving nozzle.
$u$ = peripheral velocity of blade.
$v_1$ = relative velocity of steam entering blading.
$v_2$ = relative velocity of steam leaving blading.
$V_2$ = absolute velocity of steam leaving blading.
$\alpha$ angle of jet with direction of motion.

The steam leaves the nozzle at angle $\alpha$ to the direction of motion with a velocity of $V_1$ feet per second and the blades are assumed to be moving with a linear velocity of $u$ ft. per second. Under these conditions the relative velocity of the steam jet with respect to the wheel is $v_1$, as shown in Fig. 49. In order

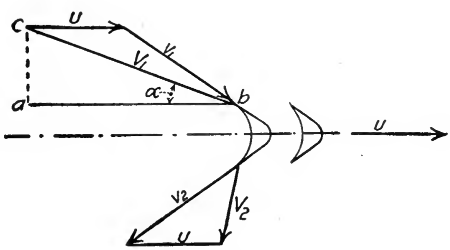



FIG. 49.—Velocity diagram of simple turbine.

that the steam may enter the blade without spattering, the blade should be made tangent to $v_1$. The steam is reversed in direction and leaves the wheel with the relative velocity $v_2$.

Much higher efficiency could be obtained if the steam entered and left the blades in the same line of direction with which the blades are moving and with a velocity equal to twice that of the blade. Under these conditions $V_2$ would be equal to zero and the maximum possible efficiency would be attained. Such an arrangement could not be used in practice because of the interference of the jet and blades, hence the steam must enter and leave at an angle $\alpha$ to the blades as shown in Fig. 49.

The absolute velocity $V_1$ can be resolved into two components as shown in Fig. 49 where $ab = V_1 \cos \alpha$ represents the useful velocity causing rotation and $ac = V_1 \sin \alpha$ causing end thrust along the shaft. In order to make $ab$ large and $ac$ small the

angle of the jet ($\alpha$) should be made as small as possible. In practice the angle ($\alpha$) is not made much less than 20°.

**99. The Simple Impulse Turbine.**—The turbine in its simplest form consists of one wheel with a series of vanes or blades on its periphery. Steam is supplied by one or more nozzles. The velocity diagram is similar to that shown in Fig. 49. The steam is reduced from full boiler pressure to back pressure in the nozzles and, upon leaving the wheel, enters the condenser or exhaust line. The steam enters the blading at high velocity, and the energy which the steam has in the form velocity upon leaving the blades is lost in the exhaust.

It can be shown that the velocity efficiency of a simple turbine is a maximum when $u = 1/2\ V_1 \cos \alpha$. As the steam leaves the nozzle with a high velocity it is necessary for the turbine to turn at extremely high r.p.m. to attain a good velocity efficiency. Simple DeLaval turbines have been built for 300 hp. and 10,000 r.p.m.; for 5 hp. and 30,000 r.p.m.

**100. Compound Turbines.**—When high initial steam pressures and low vacua are used with turbines in order to obtain a high thermal efficiency, the velocity of the steam would be extremely high if expanded from the initial to the back pressure in a single nozzle. Under these conditions the peripheral speed of the blades, and hence the r.p.m. of the shaft and propeller, would be prohibitive. If a low peripheral speed were used with a high steam velocity, the exit velocity of the steam leaving the blades would be very high and a large part of the energy in the steam would be lost in the exhaust. In order to reduce the turbine speed and still obtain a good velocity efficiency, various methods of compounding are resorted to. There are three ways of compounding impulse turbines: (1) velocity compounding; (2) pressure compounding, and (3) pressure and velocity compounding.

**101. The Velocity Compounded Turbine.**—In the velocity compounded turbine the steam is expanded through a single set of nozzles from the initial pressure $P_1$ to the condenser pressure $P_B$ and the steam is passed through alternate moving and stationary blades until the velocity has been reduced to nearly zero. Each pair of moving and stationary blades is termed a stage. The steam pressure is constant throughout the turbine casing and blading and is equal to that of the condenser.

As the steam passes through the first row of moving blades part of its kinetic energy is converted into mechanical work and the steam leaves with less velocity than it had at entrance. Upon leaving the first row of moving blades the steam enters a row of stationary blades. These serve merely to reverse the direction of the steam and, neglecting friction, there is no change in velocity as the steam passes through these blades. The steam next enters the second row of moving blades, where its velocity is still further reduced.

Figure 50 shows a velocity diagram of a two stage velocity compounded turbine. It will be observed that the blade velocity ($u$) is constant for all moving vanes and is small compared to the velocity ($V_1$) of the steam leaving the nozzle. The velocity is reduced in each stage until it has the absolute velocity of $V_4$ at exit of the last stage. A small DeLaval velocity compounded turbine is shown in Fig. 51.

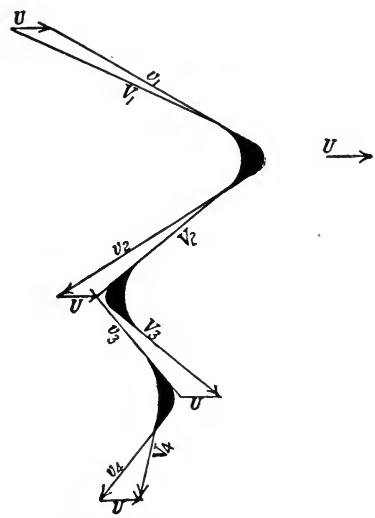


Fig. 50.—Diagram of velocity compounded turbine.

The velocity diagrams, Figs. 50 and 49, are drawn for the theoretical case when no friction exists. In the actual turbine the friction of the steam crossing the moving and stationary blades materially cuts down the exit velocity from the blades over that shown in Fig. 50. The effect of blade friction is to increase the heat contents of the steam as was described under nozzle design, and to make turbine design more complicated than the elementary case given here.

**102. The Pressure Compounded Turbine.**—In the pressure compounded turbine the steam passes successively through nozzles and moving vanes until the pressure of the steam has been reduced to that of the condenser. The nozzles are designed so that there is only a small drop in pressure and hence the velocity of the steam at the exit of the nozzles is small. The energy of the steam on leaving the first set of nozzles is partly kinetic

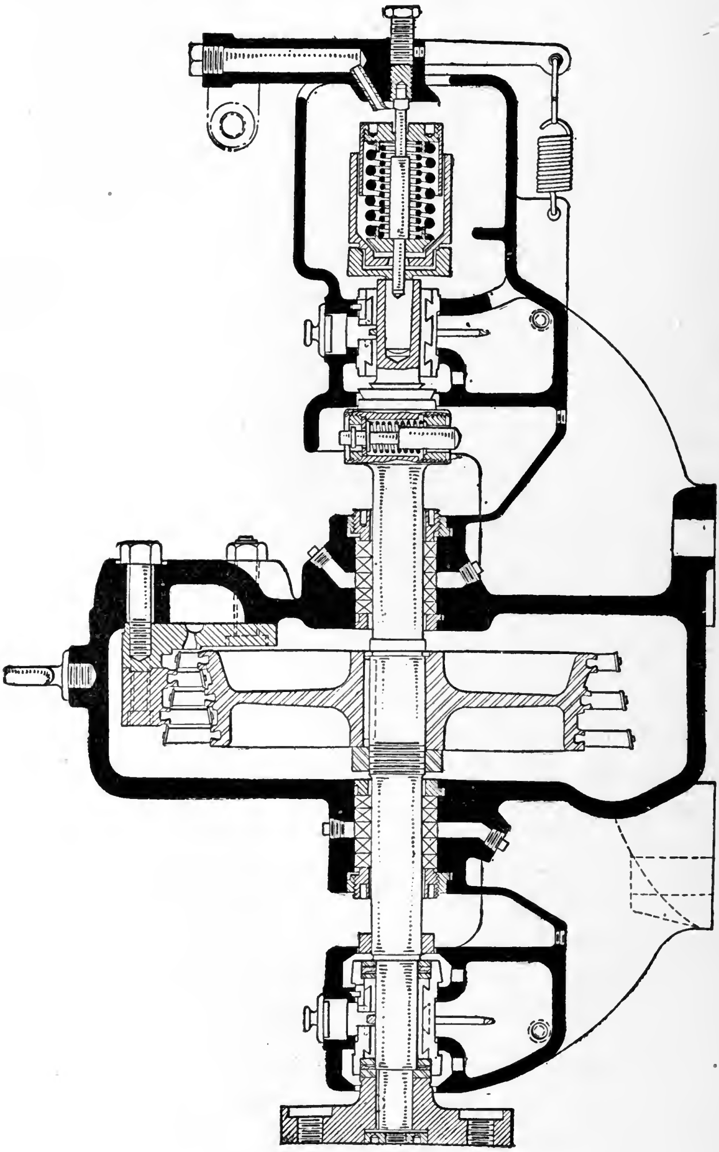


FIG. 51.—De Laval velocity compounded turbine.

(velocity) and partly potential (heat contents). The heat contents remaining in the steam is used in creating velocity in the remaining nozzles. Figure 52 shows a seven stage pressure compounded turbine. The pressure compounded turbine has but one row of moving blades per stage and the pressure varies

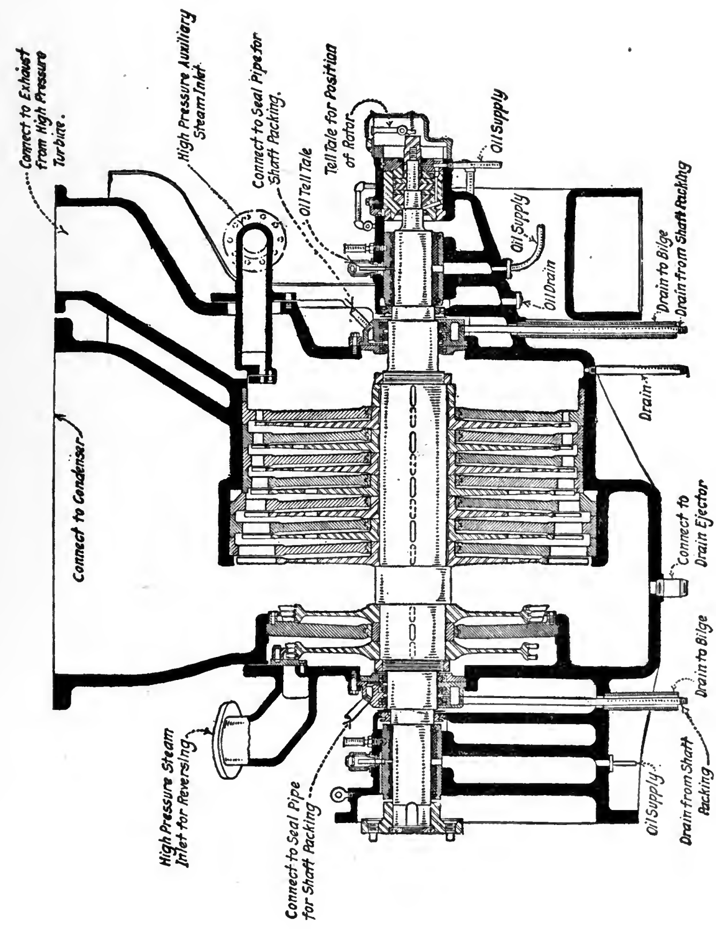



Fig. 52.—De Laval pressure compounded turbine.

from stage to stage. All the velocity acquired in the first set of nozzles is converted into mechanical work in the first row of moving blades. After leaving the first row of blades the steam enters a second set of nozzles where the velocity is again increased by a drop in pressure through the nozzles. As the steam in-

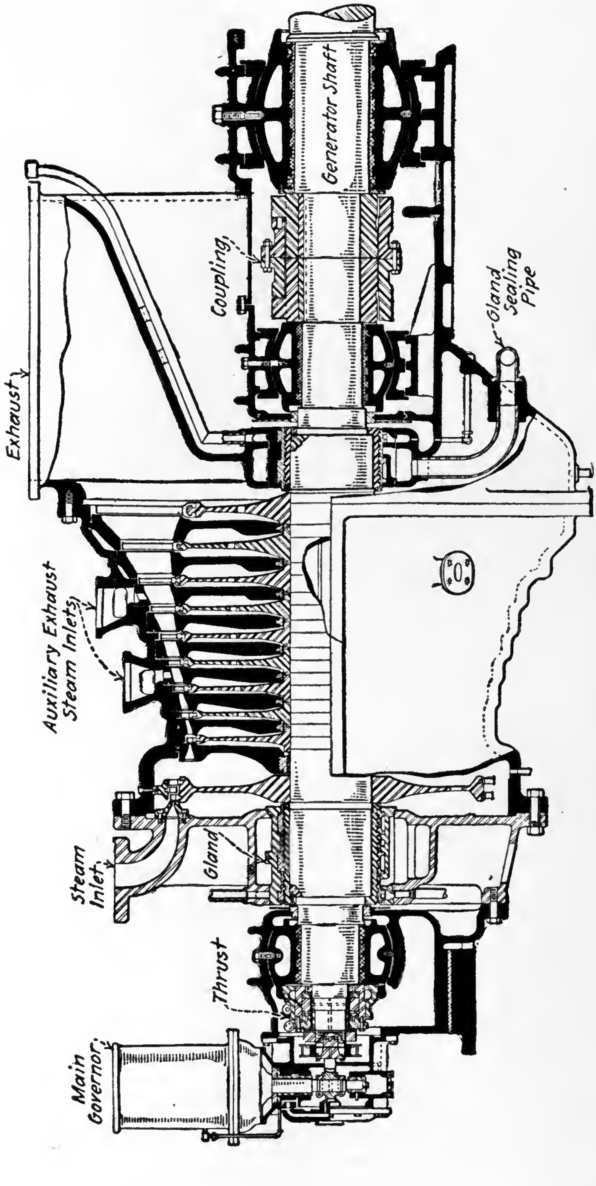



FIG. 53.—Curtis type turbine. U. S. S. "New Mexico."

creases in volume with a reduction in pressure, the area through the nozzles and blades must increase from stage as shown in Fig. 52. As the pressure in each stage is different from that in

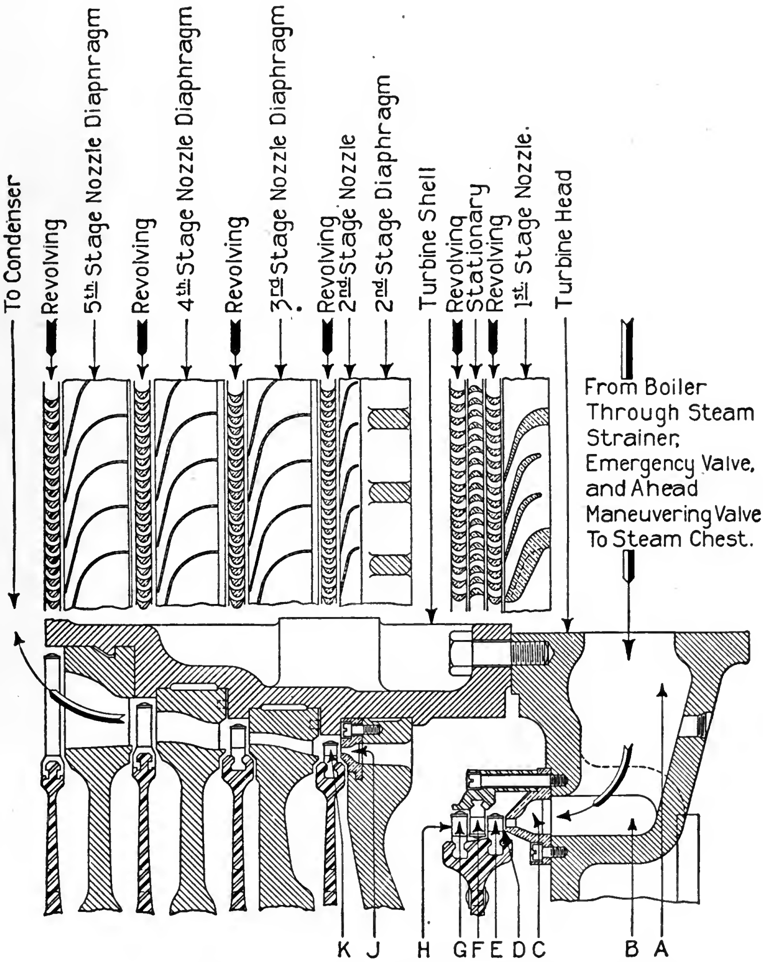



FIG. 53*a*.—Diagram of velocity-pressure compounded turbine (General Electric Co.).

the adjacent ones, each stage must be made steam tight in order to avoid leakage. This is accomplished by extending the diaphragm which holds the nozzles in to the shaft. As the circumference at the shaft is much less than it would be at the

nozzles, a tight joint is more easily made between the moving shaft and stationary diaphragm than would be possible if an attempt were made to keep a tight joint close to the nozzles. This method of securing steam tightness between stages results in the disc type of construction shown in Fig. 52.

For a turbine of $(n)$ stages the heat available per stage is $\frac{H_1-H_2}{n}$, where $H_1$ is the heat contents at the initial pressure $P_1$

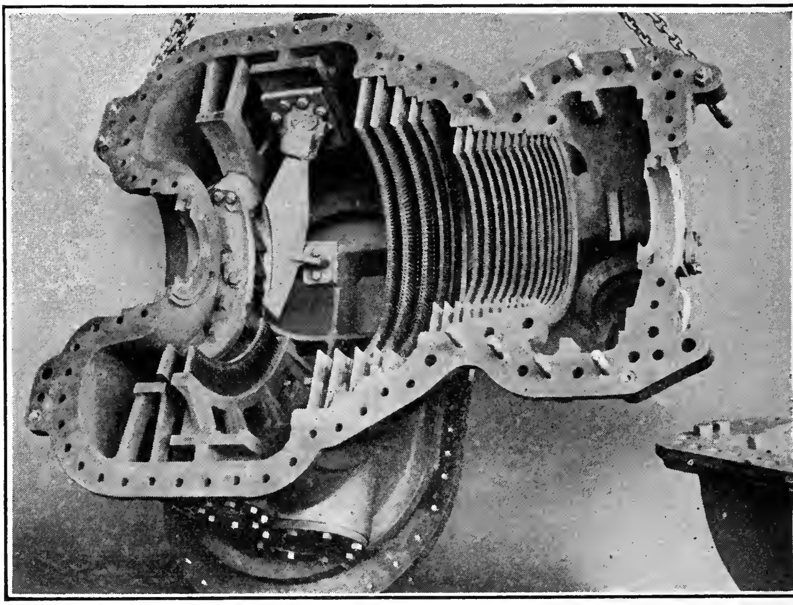


FIG. 54.—Cylinder cover. L. P. unit. Westinghouse turbine.

and $H_2$ the heat contents after an expansion to the condenser pressure $P_B$. The velocity at entrance to each stage is

$$V=224\sqrt{\frac{H_1-H_2}{n}}$$

**103. Pressure Velocity Compounded Turbine.**—This type of turbine is a combination of the two previously discussed. It consists of a number of pressure stages with several moving (velocity compounded) wheels in each stage. A turbine of this type is used where a large reduction is desired between the steam and blade velocity. Figure 53 shows a turbine with a velocity

compounded wheel in the first stage and the rest of the stages pure pressure compounded.

Figure 53a shows a diagram of the General Electric Co., Curtis type turbine (See Figs. 53 and 57). This turbine is of the pressure-velocity compound type. The first stage is velocity compounded. The steam enters through the nozzles at *C* and passes through two sets of moving blades, *E* and *G*, and one row of stationary blades *F*. The four remaining stages are of the pressure compounded type. It will be noticed that the nozzles are converging, which is always the case when the exit pressure is

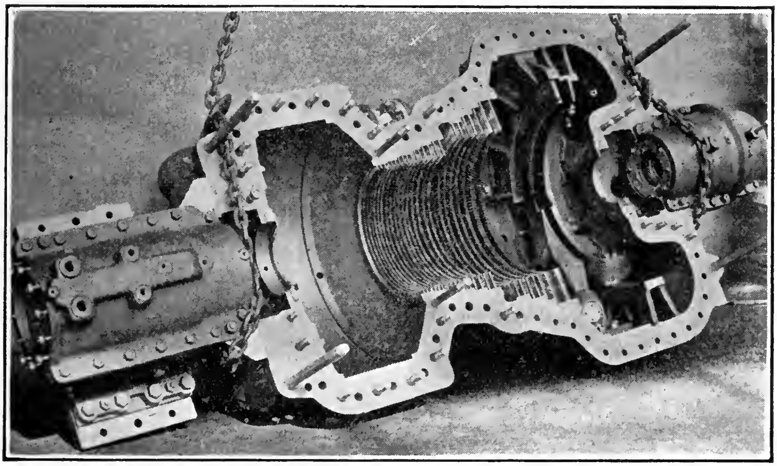


FIG. 55.—H. P. cylinder base Westinghouse turbine.

greater than .58 $P_1$. Figures 53a and 57 show clearly the disc construction of impulse turbine.

**104. The Parsons Reaction Turbine.**—The Parsons turbine uses both the impulse and reaction of the steam in doing work. The turbine consists of a large number of alternate moving and stationary rings of blading. No nozzles are used.

The steam upon leaving the throttle passes through an annular set of stationary vanes attached to the turbine casing. These blades are constructed so that the area at exit between two adjacent blades is smaller than at entrance. This causes a drop in pressure and an increase in the velocity of the steam, the same as in a converging nozzle. Upon leaving the fixed blades the steam enters the first set of moving vanes attached to the revolv-

ing rotor. Here the kinetic energy of the steam due to its velocity is converted into mechanical work and the velocity of the steam is reduced as it approaches the middle of the passage between the moving blades.

The moving blades are also constructed so that the passage between them is converging, and the steam acquires a second increase in velocity accompanied by a further drop in pressure as it leaves the moving blades. The reaction due to this increase in velocity in the moving vanes does useful work in turning the rotor. The same process takes place in each set of stationary and moving vanes along the turbine until the pressure has been reduced by successive steps to that existing in the condenser. Although the turbine is called a reaction turbine due to the reaction described above, it is in reality a combined impulse and reaction turbine, as the driving is done both by the impulse of the steam at entrance to the moving blades as well as the reaction at exit. Figures 54 and 55 show the lower casings and part of the annular rings of stationary blades of a Westinghouse-Parsons turbine. Figure 55 is the H. P. unit of a 5,500 s.h.p. cross-compound unit and Fig. 54 the low pressure unit. Figure 57 shows the rotor, cylinder base and cover of a Westinghouse-Parsons turbine. The stationary blading in the casing and the moving blades on the rotor are clearly shown in this figure. Figure 59 shows a detail of Parsons blading and the method of securing and spacing of the rotor blades.

Because of the small difference in pressure between the rows of blading in the Parsons' type of turbine the drum construction can be used as shown in Fig. 57.

This gives a stiffer rotor than the disc type of construction used in the pressure compounded turbine of the impulse type. The drum construction cannot be used with the latter type because of the larger circumferential space that would be open to leakage between the stages.

The pressure drops uniformly along the blading of a Parsons turbine, and in order to reduce leakage between successive rows of blades the radial clearance at the end of the blades is made very small.

Due to the drop in pressure throughout the turbine, the volume of the steam gradually increases from throttle to condenser. To take care of this increase in volume the blade

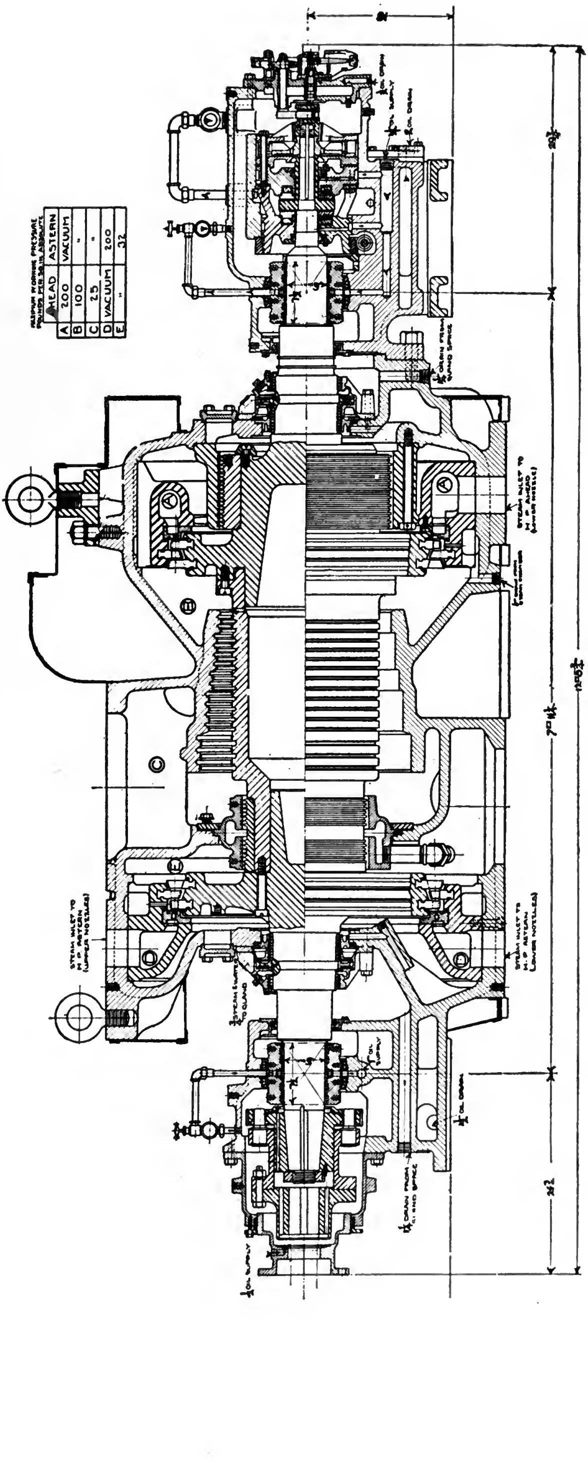



FIG. 56a.—H. P. unit Westinghouse cross-compound turbine.

heights are increased. Theoretically the increase should be of a gradual character, but in order to make a more practical construction the increase is made in successive groups as shown in Figs. 54 and 55. When the blade heights become excessive the diameter of the rotor and casing are increased (Figs. 54 and 55). This increases the circumference and also the peripheral speed of the center of the moving blades, allowing the height of blades to be decreased and yet pass the same volume of steam.

A row of fixed blades and a row of moving blades constitute a stage in the Parsons turbine.

Because of the changes in the diameter of the rotor and the action of the steam in the moving blades, an axial thrust is set up in the rotor of a reaction turbine which must be balanced by dummy pistons or by thrust bearings. The dummy pistons are shown at the right hand end of the rotors in Figs. 56a and 56b. Leakage by the dummies is prevented by labyrinth packing consisting of brass or aluminum strips on the casing fitting into grooves in the dummies. (See Figs. 56a, 56b and 57.)

The latest type of Parsons turbine used in stationary power plants has been fitted with "end tightened" blading at the high pressure end to reduce the leakage and to obviate the small radial clearance previously required with the high pressure blading. (See Art. **105.**) This system has not been used to a very large extent in marine installations but a number of ships are fitted with this blading.[1]

Dummy pistons have also been eliminated in the latest type, the axial thrust being taken care of by a Mitchell or Kinsbury type of thrust block on the rotor shaft.

The "end tightening" consists in fitting shrouding strips with side extensions around the outer circumference of both the stationary and moving blades.

The overhanging extension of these shroud rings is tapered off to a fine fin-like edge. This thin edge of the shroud ring on the stationary blades abuts against the root of the foundation ring of the following row of moving blades; and the shroud ring on the moving blades abuts against the root of the foundation ring of the preceding row of stationary blades. The axial clearances between the shroud rings and the foundation rings of the next

[1]See article and illustrations on geared turbines for S. S. "Corrientes." *Engineering* (London), vol. 110, p. 211.

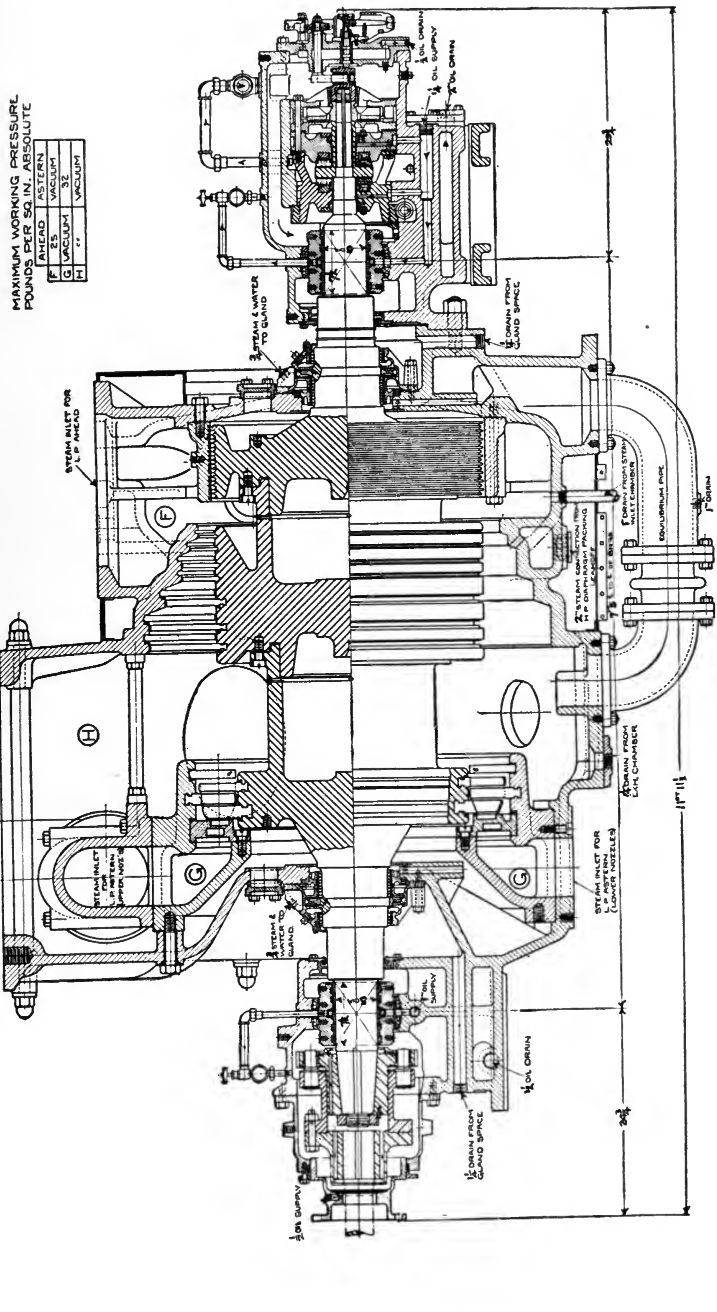



FIG. 56b.—L. P. unit Westinghouse cross-compound turbine.

set of blades can be adjusted simultaneously to any desired amount, while the turbine is running, by an adjustment at the thrust bearing at the high pressure end of the turbine.

By this arrangement the radial clearance of the blades can be made $\frac{1}{4}$ in. or larger, a high superheat can be used, and, due to the absence of dummies, leakage around the dummies has been eliminated. This "end tightening" is used only on the short

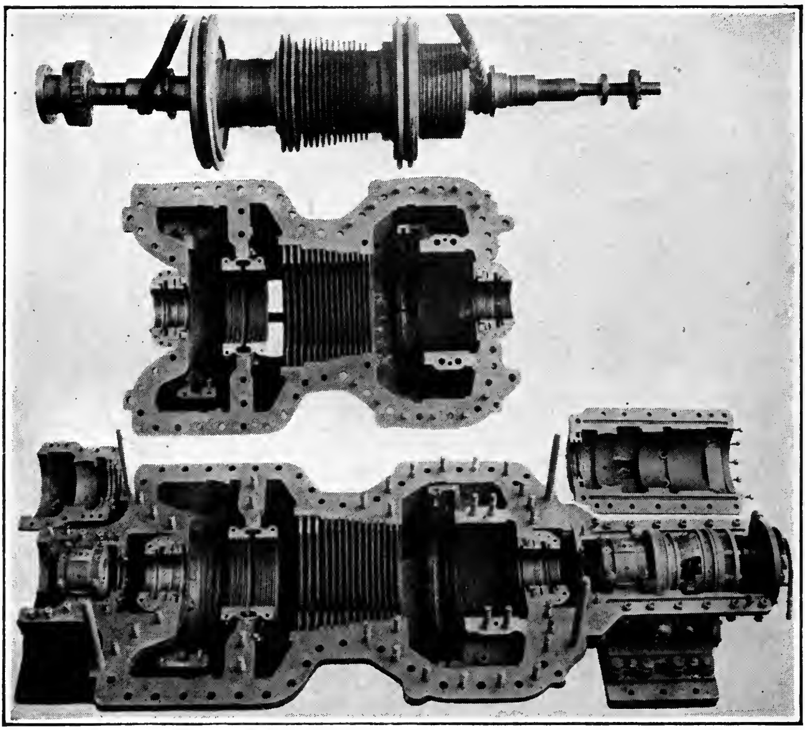


Fig. 57.—Cylinder base, cover and rotor. Westinghouse turbine.

high pressure blading where the percentage of leakage has been large when tightness was secured by small radial clearance.

As there is no change in pressure in the blading of the *impulse* turbine the radial clearance can be made as large as desired in this type, and a high degree of superheat can be used.

Test of a Parsons stationary turbine fitted with "end tightened" blades showed an efficiency ratio of .73 after 30 months operation. This turbine carried a steam temperature of 700° (275° superheat) at the throttle.

**105. Combined Parsons and Impulse Turbines.**—As already pointed out, the leakage between successive rows of blading (stages) in the Parsons turbine takes place over the end of the blades and the clearance at the end of the blades is made small to reduce this leakage. At the high pressure end of the Parsons turbine where the pressure is high the blades are extremely short because of the small specific volume of the steam. As the mechanical clearance at the end of the blades is practically the same for all blade lengths, the percentage of clearance must be very much larger with the short, high pressure blades than with the longer blades. In consequence of this, the percentage of leakage between rows of blades or stages at the high pressure end is high, resulting in a low blade efficiency.

Many turbines of the Parsons type today use a single velocity compounded impulse element with one set of nozzles at the high pressure end. A turbine of this type is shown in Fig. 56a, the nozzles and impulse blading being shown at *A*. This construction does away with a large number of rows of the inefficient, short, high pressure blading, and materially shortens the turbine, giving a shorter and stiffer rotor. The use of the impulse element also allows a higher degree of superheat to be used than when pure Parsons H.P. blading is used.

**106. Turbines for Ship Propulsion.**—All of the above mentioned types of compound turbines are used in marine work. When the turbine was first used for ship propulsion it was directly connected to the propeller shaft. This, however, was not an economic arrangement because the turbine required high blade velocity for good economy, and the propeller low r.p.m. for high efficiency. The result was that the turbines were made large in diameter to get a satisfactory relation between blade and steam velocities. The r.p.m. adopted was a compromise and was too low for good turbine efficiency and too high for good propeller performance. Because of the large diameter and low r.p.m., the turbines were extremely heavy. The "Lusitania" and "Mauretania" are examples of the direct connected turbine.

In order to have the turbine and propeller running at speeds that will give high efficiency for both, the geared turbine and electric drive have been adopted. For installations of this type in cargo vessels the turbine is run at about 3,500 r.p.m.

and the propeller at 90 r.p.m. This gives a blade velocity that is consistent with the velocity of the steam, and the turbine is light and compact. Although the propeller r.p.m. is still too high for the best efficiency for low speed cargo ships, the geared turbine and electric drive give a satisfactory solution of the problem and a propeller of nearly the maximum diameter that can be swung.

When gearing is used the turbine is generally divided into two units (cross compounded)—a H.P. unit (Figs. 55 and 56a), and a L.P. unit (Figs. 54 and 56b). The steam after passing through the H.P. unit passes across to the L.P. unit and through this unit to the condenser. The L.P. unit is in reality but a continuation of the H.P. unit, the blading being longer and diameter of the rotor larger than for the H.P. unit. Each unit is connected to the reduction gearing by a pinion, Figs. 61, 62 and 63a.

When electric drive is adopted only one turbine unit for each generator is used with the H.P. and L.P. blading on one rotor. The turbine is direct connected to a generator and runs at a speed of about 3,500 r.p.m. Figure 53 shows one of the General Electric-Curtis type velocity-pressure compounded impulse turbines used on the "New Mexico." Figure 58 shows one of the Westinghouse-Parsons turbines used on the "Tennessee." This turbine has a velocity compounded unit in the H.P. end, as explained in Art. **105**. It is of the double flow type, the steam dividing after passing through the impulse blading and the first half of the Parsons blading. Half of the steam passes over to the forward end through the cross connecting pipe, the other half continuing in the same direction as before. Exhaust connections to the condenser lead from each end of the turbine as shown.

Fig. 56a shows the high pressure element and 56b the low pressure element of a Westinghouse cross-compound geared turbine. This turbine is a combined Parsons and impulse type. The steam from the boiler enters at $A$, Fig. 56a, and passes through a set of nozzles that reduces its pressure from 200 lbs. to 100 lbs. per square inch. On leaving the nozzles, the steam enters the velocity-compound impulse element which consists of two rows of moving blades and one row of stationary blades.

At $B$ the steam enters the Parsons blading and passes suc-

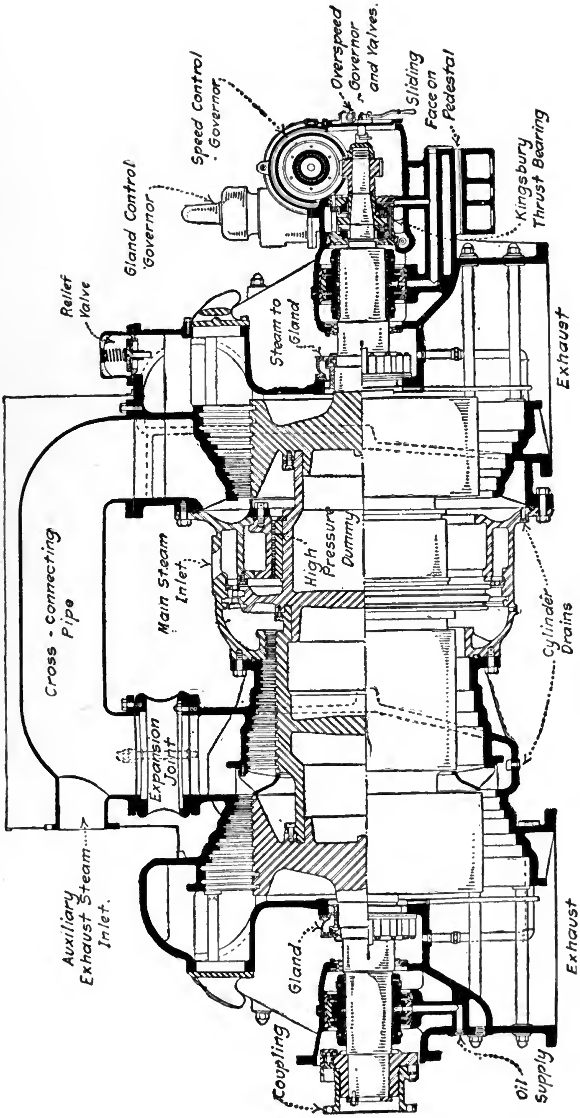



FIG. 58.—Westinghouse turbine. U. S. S. "Tennessee."

cessively through stationary and moving blades. At *C* the steam leaves the H.P. casing and passes over to the L.P. element (Fig. 56b) where it continues its expansion through the Parsons blading from *F* to *H*. At *H* the steam has been reduced to the condenser pressure and is exhausted through the exhaust trunk to the condenser.

**107. Reversing Turbines.**—As the blading in a steam turbine will allow it to operate only in one direction, all marine turbines must be fitted with a separate turbine for reversing. The only exception to this is when electric drive is used. Here the

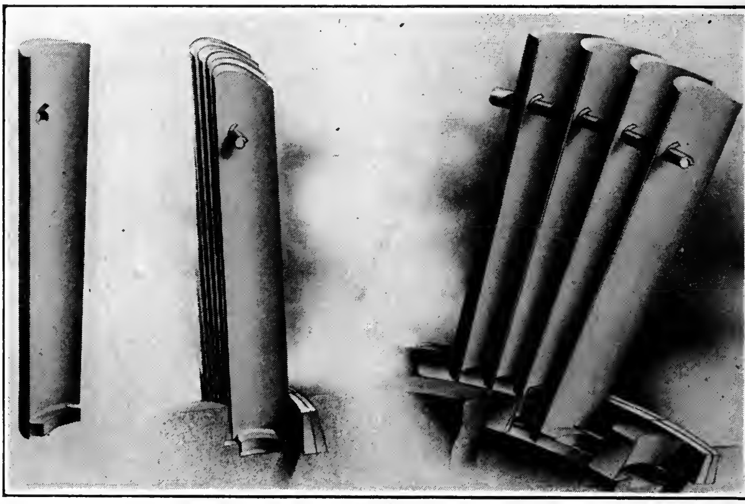


Fig. 59.—Construction and assembly of reaction blading.

turbine runs continually in one direction and the reversing is done by reversing the motor.

The reversing turbine is always placed on the same rotor and in the casing with the ahead turbine and only a small number of rows of blades are used. The astern blading is clearly shown in Figs. 56 and 60. It is placed at the L.P. end of the L.P. unit (Fig. 56b) and as the space in which it is placed is in connection with the condenser the windage loss when the turbine is running ahead is small. In the H.P. unit the astern turbine must be separated from the ahead blading by steam tight labyrinth packing as shown in Fig. 56a. When going ahead, the astern turbine in the H.P. unit is also in communication with the condenser.

For a geared turbine installation where the cross-compounded turbine is used, both units must be fitted with astern elements. The astern turbine in such a case is also cross compounded, the steam first passing through the H.P. blading and then through the L.P. blading in the other unit.

The astern turbine is very inefficient due to the small amount of blading used. Due to the low efficienny of the astern blading the steam consumption when reversing is very high and the astern power is only about 50 to 60 per cent of the ahead power. This is sufficient, however, for maneuvering. There is a small loss when going ahead due to the friction or windage loss in the astern blading.

**108. Superheat with Turbines.**—With steam turbines the flow of steam is always in one direction; hence there is no loss due to condensation caused by hot steam coming in contact with relative cold surfaces as is the case in reciprocating engines. Superheat does not have the same use as with reciprocating engines, but it is of value in increasing the initial heat contents and decreasing the loss due to the friction and windage of the steam in the high pressure end of the turbine. The lubricating oil problem mentioned in discussing the use of superheat with reciprocating engines is not present with turbines. There are objections, however, to the use of superheat with turbines that should be given careful attention.

The chief difficulty with superheat in marine turbines is the variation in temperature that comes on the reversing blading. When running ahead, the astern blading is operating in a vacuum at a low temperature, and when full steam pressure is turned into the astern turbine a large temperature change will take place in the blading which will be increased when superheated steam is used. Until recently 100° of superheat was thought to be the highest value that it was safe to carry, but the latest Cunard geared turbine ships are using 200° superheat with no apparent difficulty. When electric propulsion is used the above objection does not hold as no astern turbine is necessary.

In fixing on the steam pressure and degree of superheat to use with a turbine, full consideration should be given to the question of "heat availability" discussed in Art. **89.** A high steam pressure is more desirable than a high degree of superheat.

**109. Effect of High Vacuum.**—A consideration of the expres-

sions for the Rankine and thermal efficiency of a turbine shows that for high economy a high vacuum is desirable. This not only decreases the steam consumption by decreasing $H_2$ but also increases the horsepower of the turbine, as each pound of steam does a larger amount of work. In Art. **90** we saw that 27½ in. was the highest vacuum advisable to carry with a reciprocating engine due to the large volumes of steam to be handled. With turbines, however, the large volume occupied by steam at high vacua is not a detriment and only calls for longer blade lengths at the low pressure end. The economy of a turbine increases as the vacuum is increased up to the highest vacuum that it is possible to maintain. Vacua higher than 29 in. are possible, but, as pointed out in Art. **90,** if the vacuum is carried too high the cost of producing it will be more than the gain in economy attained. The tendency with marine installations, however, is not to carry as high a vacuum as is advisable with turbines.

The foregoing chapter gives the underlying elementary thermodynamics of the steam turbine and a description of the various types used in marine practice. For the thermodynamics of the compound turbines, losses in turbines, empirical methods of design, and more detailed information the reader should consult books devoted to the steam turbine and steam turbine design.

Details of construction and methods of operation are beyond the scope of this book; these should be acquired by actual experience and observation in the ship's engine room.

## CHAPTER XI

# THE GEARED TURBINE AND COMBINATION MACHINERY

**110. General.**—As pointed out in the last chapter, mechanical gearing has been interposed between the turbine and the propeller in order to improve the efficiency of both propeller and turbine by allowing each to run at the r.p.m. for best efficiency. The introduction of the geared turbine is very recent and its development has been extremely rapid. The first installations of the geared turbine were in 1910 on the U.S.S. "Neptune" (Westinghouse gears) and the S.S. "Vespasian" (Parsons gears). In 1922 more than a thousand merchant vessels and a like number of naval craft have been fitted with this type of propelling machinery.

The development of the geared turbine has been so rapid that many failures have resulted due to lack of experience, poor judgment in design, inferior workmanship, and mishandling on the part of the crews. During the World War (1914-1918) the demand for geared turbine sets was so pressing that many firms started building these units without proper experience of the service conditions under which they would be used. As the speed of production was very rapid a large number of sets were installed before the sea trials of the first set had taken place. The result of this has been that many shipowners hesitate to adopt this type of machinery and the geared turbine has continued to suffer from these early and natural mistakes.

At the present writing (1922) the geared turbine is an unqualified success. Experience with the unit in service at sea both by designers and operating engineers has remedied all the earlier mistakes and the unit is being installed without hesitation by many engineers and shipowners. A number of ships with geared turbines are now in operation after completing from 120,000 to 200,000 miles without gear troubles.

**111. Types of Gearing.**—Reduction gears are built in both

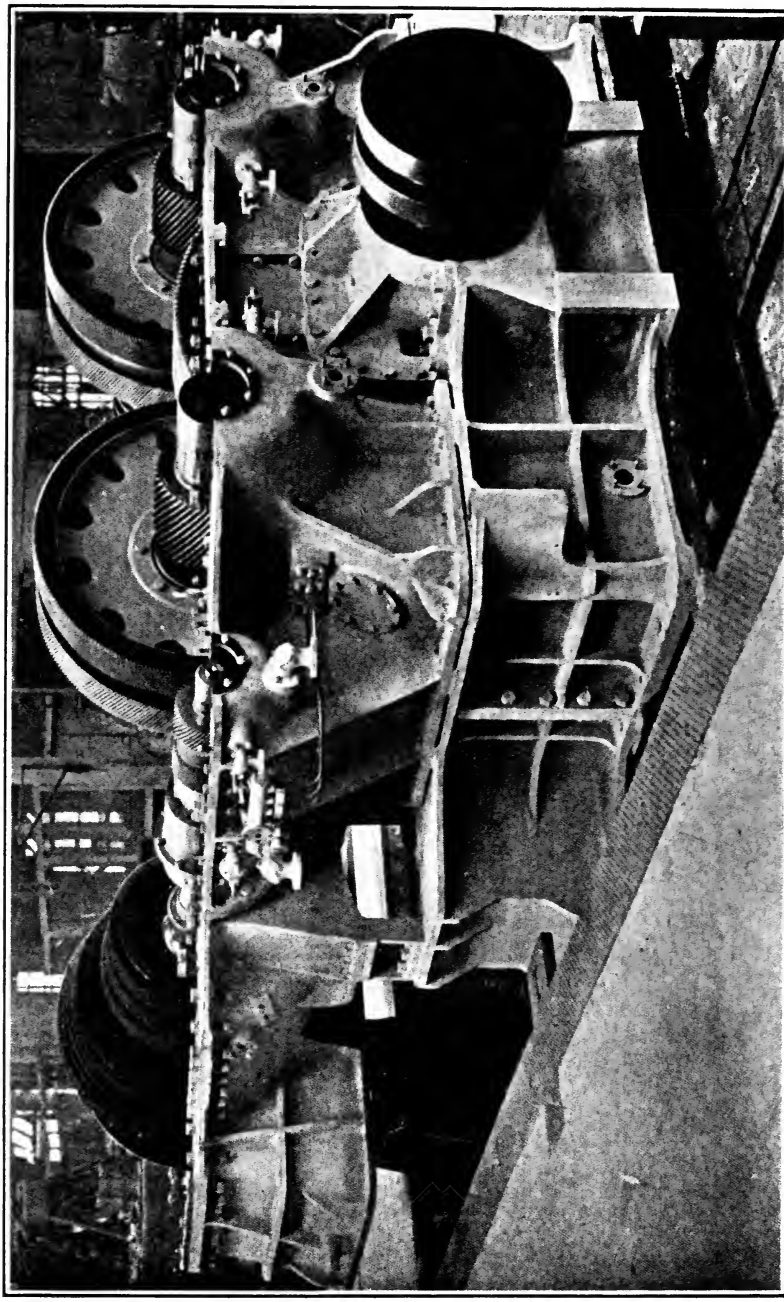


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FIG. 61.—Double-reduction gearing and L. P. turbine unit.

single and double reduction units, depending upon the propeller speed desired. Single reduction units, consisting of one set of gears, reduce the revolutions from 1:15 to 1:20, and double reduction units, consisting of a second pinion and gear, reduce the r.p.m. about 1:36.

Most naval vessels, where high propeller speeds are permissible, use single reduction gearing. Figure 62 shows a single reduction unit with cross-compound turbines. A number of passenger

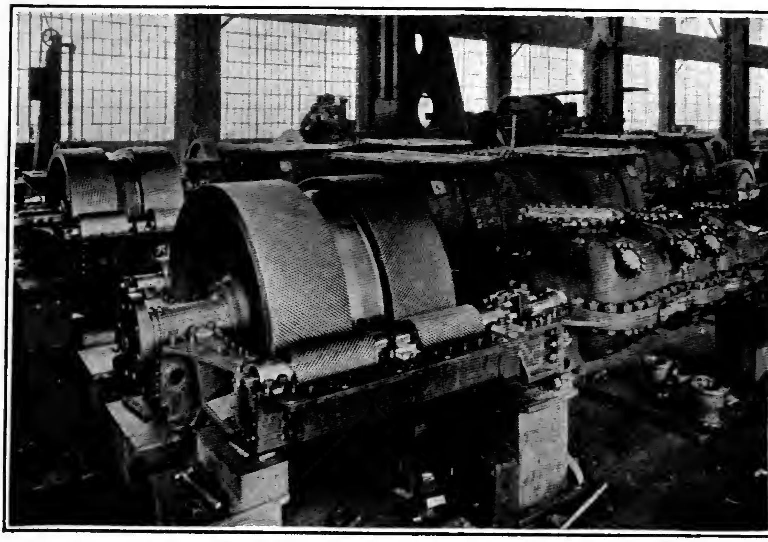

FIG. 62.—Single-reduction gears for destroyers.

vessels, cross channel steamers, and a few cargo ships have also been fitted with single reduction sets. For these ships the propeller speed is in the neighborhood of 125 r.p.m. and the reduction ratio is about 1:18. This small reduction, however, does not give the most advantageous turbine speed except for powers between 10,000 and 20,000 s.h.p. In order to get the lowest steam consumption possible, slow speed cargo ships of low power should always use high turbine speed and double reduction gearing.

The present turbine speed of 3,600 r.p.m., used with double reduction gears, is still too low for best economy with turbines of between 2,500 and 3,500 s.h.p. The present turbine speed

of 3,600 r.p.m., used with double reduction gears, gives the maximum economy for powers between 4,000 and 8,000 s.h.p. For higher horsepowers, however, the maximum turbine efficiency is obtained at lower revolutions.

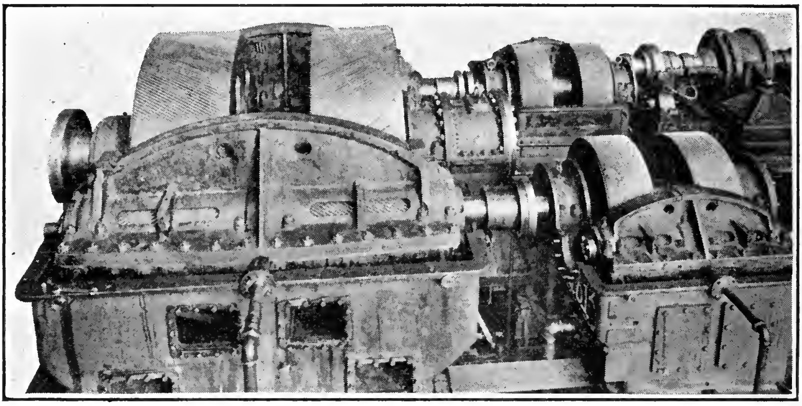

FIG. 63.—Westinghouse three case double-reduction gearing.

Figure 61 shows a double reduction set of the one plane type with cross-compound turbines as used on the latest Cunard vessels. The two sets of gears are clearly shown in this figure.

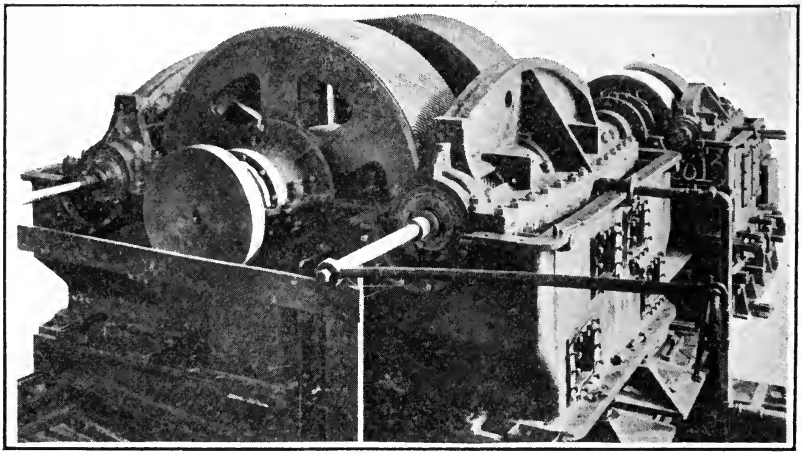

FIG. 63a.—Low speed pinion and gear of set shown in Fig. 63.

Gears are built with both fixed or rigid bearings and with a flexible floating frame. The latter type is built exclusively by

the Westinghouse Elec. & Mfg. Co. under the Melville-Macalpine patents. All other gear builders use the rigid bearing type. Figures 63 and 63a show views of a Westinghouse reduction gear built for S. S. "Manalani" and "Manukai" of the Matson Navigation Co. These gears are for a cross-compound installation of 5,500 s.h.p. The high speed pinion (enclosed in the floating frame) and the high speed gears are shown at the right in Fig. 63, and the low speed pinion and gear at the left. This gear is of the three case type. The floating frame (Fig. 64), which carries the driving pinion and its bearings, is independent of the housing except for one support at the center. This arrangement automatically maintains the alignment of the pinions and gives

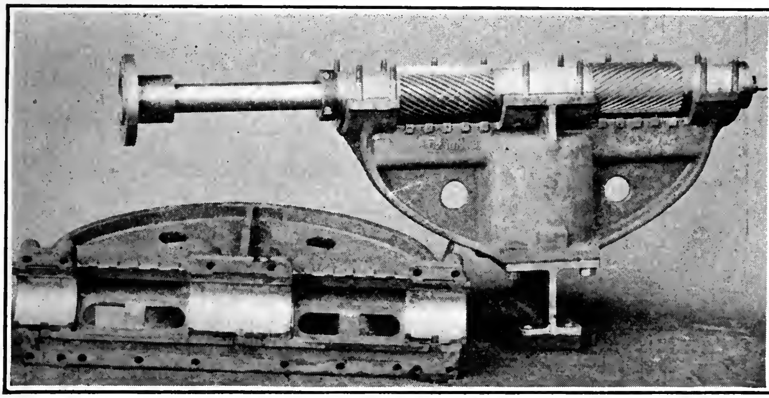

Fig. 64.—High speed pinion and floating frame.

a uniform distribution of tooth pressure over the tooth face.

Both the fixed bearing and floating frame types of reduction gearing are giving excellent results in service on shipboard. Data on a few typical installations are given in Table X.

The turbines described in Art. **106** and illustrated in Figs. 54, 55, 56a, 56b, 57 and 60 are used with the gear drive.

Figure 60 shows the General Electric Company's type of geared turbine. This turbine consists of a single unit in contrast to the cross-compound turbines illustrated in the other figures. The astern blading is shown at the left hand end of the turbine.

A recent novel installation of a Parsons geared turbine on the S. S. "Corrientes"[1] consists of three units; the H.P. and I.P.

[1]See *Engineering* (London), vol. 110, p. 211.

units are geared to one wheel and the L.P. unit gears with a second wheel on the opposite side of the propeller shaft from the H.P. and I.H. units.

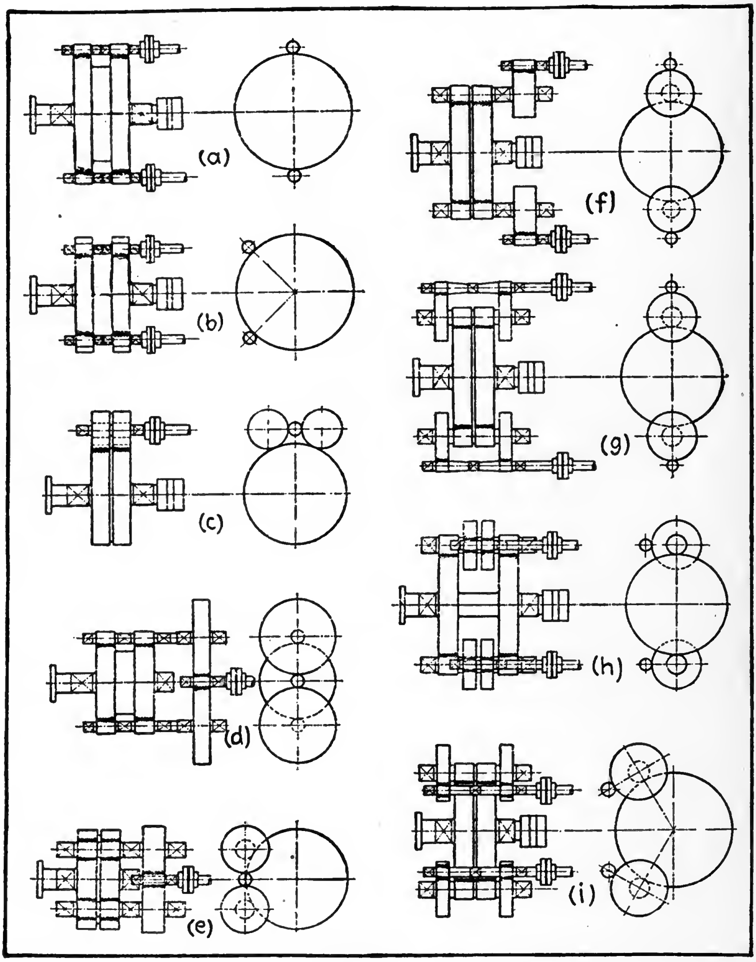


FIG. 65.—Gear arrangements.

**112. Gear Arrangements.**—Various arrangements are used for reduction gearing, depending on the type of turbine drive adopted. Figure 65 (taken from a paper by Robert Warriner, "Reduction Gears for Ship Propulsion," before the Society of Naval Architects and Marine Engineers) shows the outlines of

a number of gear arrangements for both single and double reduction. Arrangements c, d, and e are adopted when only one turbine unit is used and the other arrangements are of the two pinion type used with cross-compounded turbines. The double reduction arrangements shown in g, h and i require only one gear casing for the gearing while arrangement f requires three gear casings. This latter arrangement is used on tankers where the machinery is located aft. The turbines are well separated so that the propeller shaft can be readily withdrawn without disturbing the turbines. The three case arrangement is also used for large units as shown in Fig. 63a.

**113. Gear Construction.**—Because of the high pitch line velocity all reduction gears are of the double herringbone or helical type (Fig. 62), the teeth being cut right-handed on one gear and left-handed on the other to balance the end thrust. The involute tooth is used and the angle is generally between 22 and 45°. Tooth pressures are much less than formerly used and run between 300 and 450 lbs. per inch of face for the high speed elements and between 600 and 800 lbs. for the low speed elements. The gear wheel consists of a cast iron spider or built-up forging with a forged steel rim shrunk on to form the face for cutting the teeth.

Double reduction gears are built in the one plane and two plane types. In the one plane type the center line of the turbine and propeller shafts are in the same horizontal plane as shown in Figs. 65a, d, f and g; in the two plane type the turbine shafts are in a plane above that of the propeller shaft as shown in Figs. 65b, c and h.

In the two plane type the low speed pinions are in contact with the low speed gear near its top (Fig. 65e). This allows an increase in the diameter of the low speed gear without necessitating an increase in the diameter of the high speed gear and a consequent increase in the distance between gear centers. A large diameter of the low speed gear is advisable in order to keep the tooth pressures down without unduly increasing the width of the gear face. (See Fig. 60.)

**114. Requirements for Successful Operation.**—In order to avoid failure with geared turbine the utmost care must be given to material, design, construction, installation and operation. As already pointed out, the design of the gearing has now been

perfected and, with the low tooth pressures used, failures due to faulty design are now practically eliminated. Gears that have successfully passed the shop tests have sometimes failed, however, when placed in service on shipboard. The reasons for the more severe conditions in service as compared to shop test have now been pretty well investigated and the results of these investigations are worthy of attention.

Tests on the S.S. "Jebsen"[1] showed that the torque varied for a 2,000 s.h.p. unit from zero to 3,500 s.h.p. (75 per cent overload) in 3 seconds and variations of less magnitude occurred very rapidly over wide ranges under certain conditions. A paper by R. J. Walker before the Inst. of Naval Architects in 1921 deals at length with the variation in torque in geared turbine installations. An examination of the gears of the "San Florentino" and "San Fernando," in which the turbines were placed aft, showed indications of excessive pressure at definite angular positions around the gear. This was shown conclusively to be caused by variations in the resistance of the propeller blade due to the variations in intensity of the wake. The wake intensity varied over the propeller disc area, being a maximum at the center line of the stern post and close to the surface. In these ships, the frequency of the four blades of the propeller passing the stern post also synchronized with the natural period of torsional vibrations of the shaft. The synchronism was destroyed by fitting a three-bladed propeller. A careful study of the variation in the torque by means of the torsion meter showed variations of 30 per cent each side of the mean value, a maximum and minimum occurring four times a revolution. For a propeller running at 90 r.p.m. this would give variations in torque of about 60 per cent twelve times a second.

From the foregoing it is seen that care must be exercised that the natural period of torsional vibration of the shaft, fixed by the amount of inertia of the propeller and gearing and flexibility of the shaft, does not synchronize with the variation in propeller wake.

The variation in the propeller load occurs as the blades pass through the strong wake current at the stern post; hence the frequency becomes four times the r.p.m. for a four-bladed propeller and three times the r.p.m. for a three-bladed propeller.

[1] *General Electric Review*, Feb., 1921.

Gears fitted aft have been found to wear more quickly than those placed amidships due to the shorter length of shaft between gears and propeller. The long flexible line shafting used when the turbine is installed amidship dampens out some of the variation in torque caused by the irregular action of the propeller.

The successful operation of gears calls for exact attention to lubrication. Not only must a regular and adequate supply of lubricating oil be supplied to the gears and bearings but the oil supply must be kept free from grit, lint, foreign matter and all traces of salt water.

**115. Lubricating Oil System.**—Three types of lubricating oil systems are used for lubricating the gears and turbine bearings: the pressure system, the gravity system, and the combined pressure-gravity system. In the pressure system the oil is supplied direct to the gears and bearings by the lubricating oil pump. In the gravity system the pump delivers the oil to overhead tanks and the oil flows from the tanks to the gearing by gravity. In the pressure-gravity system oil is supplied by the oil pump direct to the bearings and gears as in the pressure system. The pumps operate, however, at a higher rate than necessary and the excess oil passes through a relief valve to an overhead gravity tank. If for any reason the pressure falls, the supply in the gravity tank automatically flows to the gears and bearings, thus insuring a temporary supply until the pump can be started or the turbine stopped. The former system is used almost universally on naval vessels as there is not sufficient height to install a gravity system. The gravity system is by far the best system and is used with practically all geared turbine installations on merchant ships.

A gravity lubricating oiling system used by the De Laval Steam Turbine Co. is shown in Fig. 66, and one used with Westinghouse gears in Fig. 67. This is typical of all gravity systems. The advantage of the gravity system is that in case of any accident to the pump the oil supply in the gravity tanks is sufficient to last several minutes, giving the engineer on watch time to shut down the turbine before the oil supply fails.

As shown in Fig. 66 the oil from the gear casing and turbine bearings drains to a sump tank below the engine room flooring. From here it is pumped by the lubricating oil pump to the overhead gravity tanks. An oil cooler is interposed in the

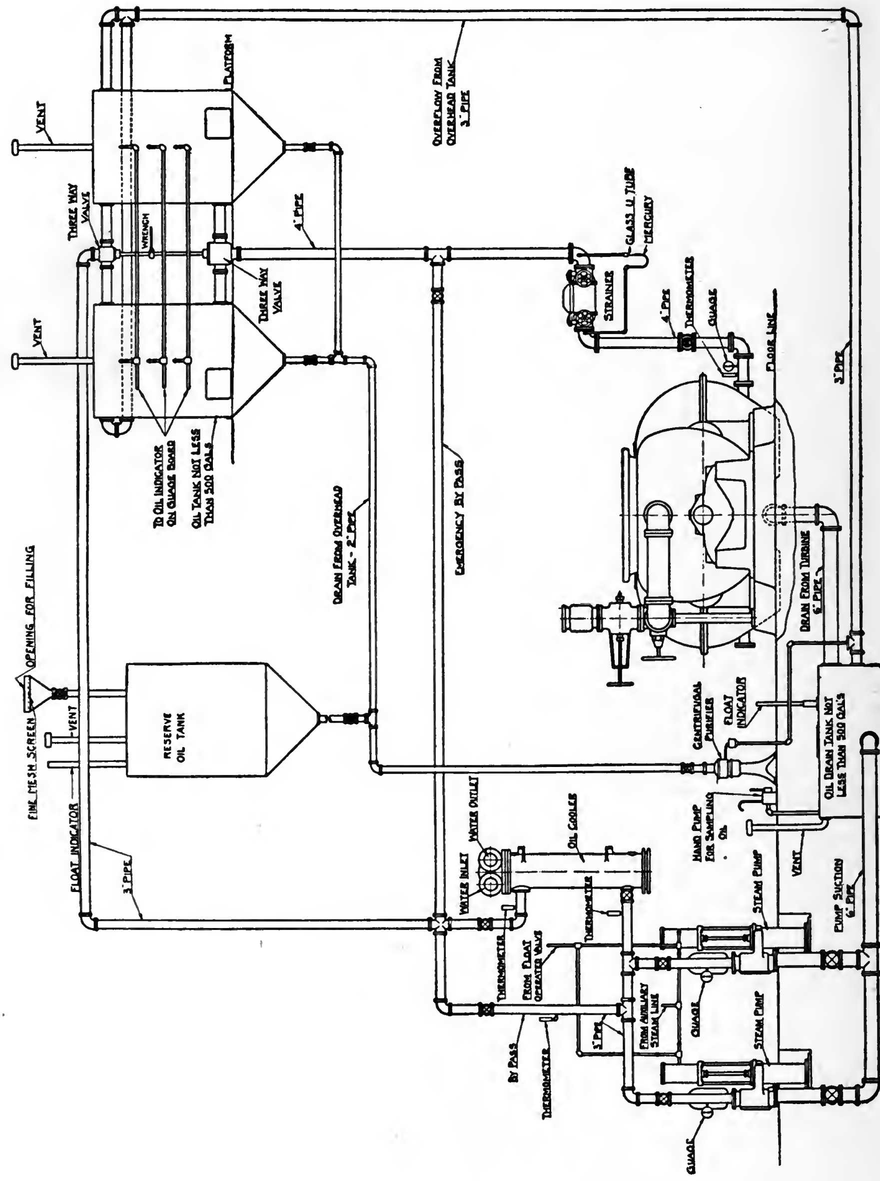


FIG. 66.—Lubricating oil system for De Laval geared turbine.

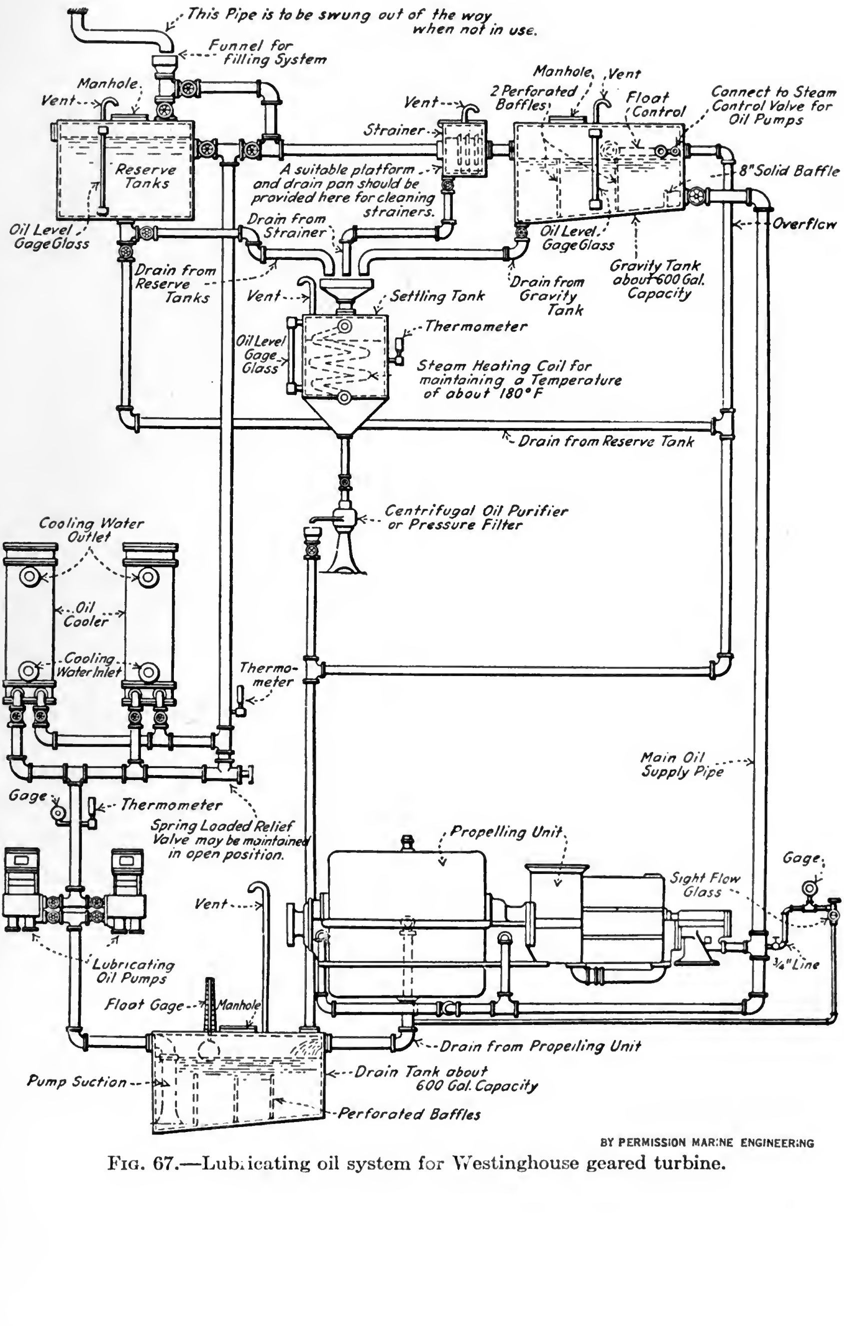


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FIG. 67.—Lubricating oil system for Westinghouse geared turbine.

discharge of the pump to remove the heat absorbed by the oil in its passage through the gears and bearings. The oil supply to the turbine is withdrawn from the gravity tanks at a point somewhat above the bottom. This allows all water and sediment to settle to the bottom of the tank where it can be drawn off to the oil separator. The arrangement is clearly shown in Fig. 66. Centrifugal purifiers are installed in all lubricating oil systems for removing water and foreign matter from the oil. If used while underway a large part of the foreign matter will settle out. In port the whole oil supply can be put through the purifier in a short time.

When the pressure system is used, the throttle valve to the turbine should be designed to be held open by the oil pressure. With this device a failure of the oil pump or oil supply would immediately cause the throttle to close before any serious damage could result. Electric alarms should be fitted on all systems to notify the engineer of any falling off in the oil supply.

**116. Essential Requirements of an Oiling System.**—The oiling system is by far the most important auxiliary installed in a geared turbine ship and it is essential that in design and operation it be the best possible to obtain. The requirements for an oiling system may be summed up as follows:

*1. Oil temperature.*—The same supply of lubricating oil is used for both gears and bearings. The duty of the oil is to carry away the heat generated by friction as well as to lubricate; hence the oil leaves at a temperature higher than that at which it entered. Gears operate best with the oil, entering at a low temperature, while bearings wear longer with an oil of higher temperature and lower viscosity. If oil of high viscosity were used for the gears the spray nozzles would not function and the pumping of the oil would involve difficulties; hence the viscosity of the oil must be kept low. The temperature used for the oil entering the bearings and gears is between 90 and 120°F. The oil should have a Saybolt viscosity between 300 and 500 seconds at 100°F. The oil supply should be sufficient to carry away the heat generated (about 5 per cent of the shaft horse power) with a rise in temperature of 10 or 20°F. Oil coolers are fitted for reducing the temperature back to the proper value for entering the gears and bearings, i.e., between 100 and

110°F. When operating in cold water the cooler may often be by-passed. It is advisable to fit coolers in duplicate.

*2. Oil pressure.*—The gravity tanks should be high enough to give a pressure at the gear nozzles and bearings of between 6 and 7 lbs. per square inch. To obtain this pressure and to give sufficient head to overcome the losses in valves, strainers and pipes, the overflow of the gravity tank should be located between 25 and 30 feet above the center line of the shaft.

*3. Oil supply.*—The practice with geared turbines is to make the capacity of the gravity tanks sufficient to hold about four or five minutes' supply. With powers between 2,500 and 3,500 s.h.p. and 10° rise in temperature this will require two 500 or 600 gallon tanks. A third tank (see Fig. 66) of about the same capacity is fitted as a reserve tank. The sump or drain tank below the turbine is given a capacity of between 500 and 1,000 gallons for horsepowers mentioned above. For a reasonably tight system, leakage should not exceed a gallon per day.

*4. Cleanliness of Oil.*—It is very essential that the oil supply be kept absolutely clean; and dirt, sediment, water and lint be kept out of the system. Filters and strainers should be installed in the system, as well as a centrifugal purifier (Figs. 66 and 67) to purify the oil that settles in the bottom of the gravity tanks. As water, and especially salt water, is disastrous, it is advisable to have the oil pressure in the coolers greater than that of the salt circulating water, for then any leakage that occurs will be a leakage of oil overboard instead of salt water into the system.

**117. Combination Machinery.**—This type of machinery consists of a triple screw arrangement with two triple expansion engines on the wing shafts exhausting to a low-pressure turbine on the center shaft. It is an excellent arrangement on account of the high efficiency of the reciprocating engines at high pressures and the high efficiency of the turbine at low pressures.

The low pressure end of the triple expansion engine is very inefficient due to the large volume of steam handled, high cylinder condensation, friction and leakage. The high pressure end, however, is very efficient if a small degree of superheat is used to eliminate the cylinder condensation up to cut-off. The high pressure end of the turbine, on the other hand, is not so efficient as the low pressure end, due to the greater friction

of high pressure steam. As pointed out in Art. **109,** the L.P. turbine can use a high vacuum to good advantage, increasing the horsepower and economy thereby.

The disadvantages of the installation are its bulkiness and large fore and aft space required to install the center line L.P. turbine aft of the reciprocating wing engines. If the three units can be installed abreast this disadvantage does not exist. Unless a high vacuum is carried and the L.P. turbine bladed for the highest possible vacuum, the gain in economy will not be realized. The weight and space occupied is greater than that of the geared turbine.

Combination machinery makes an ideal installation for low r.p.m. and the economy is as good as that of the high speed turbine and will give a substantial increase in power over that possible with triple expansion engines alone. Except for the introduction of the geared turbine this type of machinery would have been used to a larger extent.

Combination machinery has been fitted on the "Otaki," "Rochambeau," "Olympic," and "Minnekahda."

**118. The Combination Machinery Installation on S.S. "Otaki."**—This was the first installation of this type of machinery and as sister ships, the S.S. "Onari" and S.S. "Oparva," were fitted with twin reciprocating engines, comparisons in service are available. The "Otaki" was lengthened 4 ft. 6 in. over her sister ships to make up for the cargo space lost by the third shaft alley. Otherwise the hulls were identical. It developed on the trials that the reduction in fuel consumption and reduced size of boilers required on the "Otaki" were more than enough to compensate for the space lost by the shaft alley.

The principal characteristics of the "Otaki" are:

| | |
|---|---|
| L.B.P. = 464 | Displacement = 11,700 tons |
| D.w. = 9,000 tons | Draft = 27 ft. 6 in. |

Speed = 12 knots fully loaded; 14 knots with 5,000 tons d.w.

In going astern only the reciprocating engines are used; hence the turbine is not equipped with astern blading. Two condensers are fitted so that when reversing each reciprocating engine can exhaust to a separate condenser. The ratio of the H.P. and L.P. cylinders is 1:5.6 and on the sister ships the ratio is 1:7.93. The boiler pressure used was 196 lbs. abs. and the

vacuum was 28.2 in. The L.P. cylinder exhaust was 11.5 lbs. absolute and the initial pressure at the entrance of the turbine was 9.8 lbs. abs. One-third of the total power was developed in each unit. The steam consumption on the "Otaki" was 13.7 lbs. per I.H.P. per hour against 16.5 lbs. on the "Onari." The machinery weight of the "Otaki" is only 3.25 per cent greater than that of the sister ships.

The real comparison, of course, is in the results obtained in service, where it was found that the fuel consumption of the "Otaki" with combination machinery averaged between 12 and 15 per cent less than used on the sister ships with twin triple expansion engines.

TABLE X. TYPICAL REDUCTION GEARS

| Ship | Type of gear | S.H.P. per shaft | Propeller r.p.m. | Reduction ratio | Diam. of of gear, inches | Face width of gear, inches | Arrangement |
|---|---|---|---|---|---|---|---|
| U.S.S. "Wadsworth".... | Single | 8,000 | 450 | 5.56<br>3.36 | 66.47 | ....... | Cross comp. |
| H.M.S. "Hood"........ | Single | 36,000 | 210 | 7.10<br>5.20 | 143.8 | 75 | Cross comp. |
| H.M.S. "Seafire"....... | Single | 14,000 | 350 | 8.55<br>6.50 | 85.9 | 40 | Cross comp. |
| U.S. T.B.D............. | Single | 13,500 | 455 | 5.95 | ........ | ....... | Cross comp. |
| S.S. "T.P. Beal"........ | Single | 2,200 | 75 | 21.40 | 130 | 32 | Cross comp. |
| Passenger ship.......... | Single | 6,000 | 125 | 14.3 | 143 | 38 | |
| H.M.S. "Dauntless".... | Single | 20,000 | 275 | 6.55 | 99.9 | 58.25 | Cross comp. |
| S.S. "Sibboney"........ | Single | 4,500 | 120 | 12.5 | 106.75 | 48 | Cross comp. |
| | Double | 4,000 | 90 | 38.8 | H 57.9<br>L 54.75 | H 19.6<br>L 48 | Single Turbine |
| | Double | 2,500 | 90 | 38.8 | H 55.6<br>L 52.75 | H 18<br>L 45 | Single turbine |
| S.S. "Robin Gray"...... | Double | 3,000 | 100 | 30.0 | ........ | ....... | 2 plane gears<br>Single turbine |
| S.S. "Scythia".......... | Double | 6,250 | 80 | H 40.0<br>L 22.7 | ........ | ....... | Cross comp. |

# CHAPTER XII

## THE ELECTRIC DRIVE

The electric drive is a recognized form of ship propulsion and has proven itself beyond all doubt as a reliable and economical type. Several battleships and a number of cargo vessels and smaller craft have been fitted with electric drive and have been highly successful in service.

**119. Advantages.**—The outstanding advantages of this type of propulsion are: flexibility of control, large reversing torque, and, in some installations, high economy at reduced speeds. Due to these features, it is the ideal machinery for battleships and other types of ships where ease of maneuvering and high economy at low speeds are essential. Its suitability for merchant ships is a much debated question today. In space occupied, weight, cost, and economy at full power, the electric drive has no advantages over the geared turbine. Because of its excessive weight and large space required, electric propulsion is not applicable to high speed ships. A full comparison of this type of propulsion with the other standard types is given in Chap. XIV.

While the design and selection of the types of motors and generators and the installation of the electrical machinery and its controls belong in the field of the electrical engineer, it is highly essential that the marine engineer and naval architect be thoroughly conversant with this important type of propelling machinery.

**120. Essential Features.**—The electric drive is essentially a form of reduction gear. The current is generated by a two-pole, high speed turbine driven generator and is transmitted to a low speed, multi-pole motor on the propeller shaft. The steam turbine that drives the generator runs at speeds between 2,000 and 3,500 r.p.m., which is practically the same speed as used with geared turbine installations. This gives a small,

light weight turbine of high efficiency that is ideal for use on shipboard.

In order to allow the use of a light, high speed generator, alternating current of high voltage has been used in all installations of any appreciable power. After further developments of high speed direct current generators of high voltage, this type of machinery may be introduced.

The revolutions of the propeller, and hence the speed of the ship, are controlled as follows:

1. By varying the speed of the turbine and thus changing the frequency of the current.
2. By special motor windings whereby the number of poles may be varied.
3. By shutting down one of the generators when more than one is installed.
4. By inserting resistance in the rotor circuit, when the induction motor is used.

**121. Types of Propelling Motors.**—Several different types of motors have been advocated for ship propulsion and practically all of these have been installed and are now in service. Each type of motor has its own advocates and each has some merits and advantages not possessed by the others.

The types of motors used for ships populsion are:

1. Ordinary wound secondary induction motor.
2. Squirrel cage motor.
3. Combined squirrel cage and wound secondary.
4. Synchronous motor.

Diagrammatic schemes of all the above types are shown in Figs. 68, 69, 70 and 71.[1]

The induction motor with secondary control inserted in the rotor circuit has been used on some electrically propelled ships. This gives a motor with good torque characteristics for reversing and is used where a large amount of quick reversing is necessary. On cargo ships the inserted resistance is used to give a high torque in starting and reversing and is short circuited at full speed; on the U. S. S. "Tennessee" the secondary windings have

[1]These figures have been taken from a paper by W. E. Thau before the Society of Naval Architects and Marine Engineers, 1921. This paper gives an excellent discussion of electric propulsion and has been drawn from in preparing this chapter.

a variable control and are used with liquid rheostats for controlling the speed of the motor.

The synchronous motor appears to be in favor for merchant ship drives. Where only a small amount of maneuvering is necessary this is probably an ideal motor. For maneuvering,

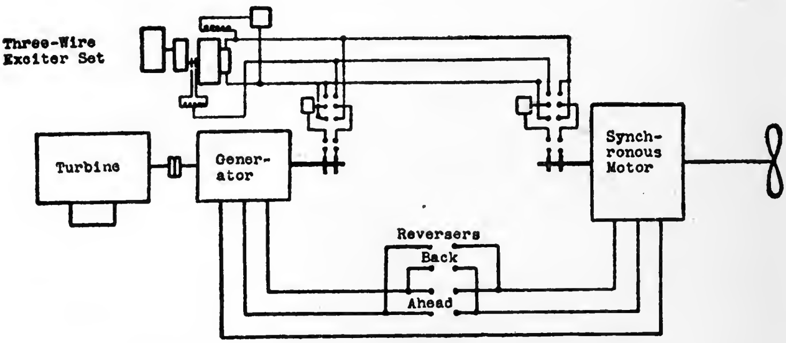



Fig. 68.—Diagram of synchronous motor installation.

the motor is operated as an induction motor and is built with a substantial squirrel cage winding. The synchronous motor is more efficient than the induction motor and has a unity power factor against a power factor of .70 to .80 for the induction motor. The synchronous motor is appreciably lighter and

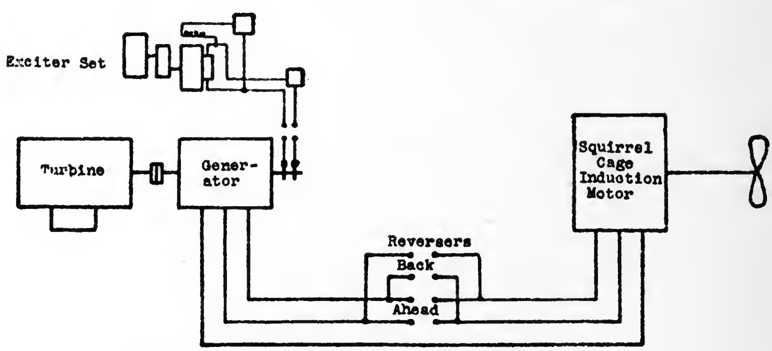



Fig. 69.—Diagram of squirrel cage motor installation.

shorter than the induction motor and its generator is slightly smaller and lighter. It has a larger air gap and is more easily repaired. The control, however, is more complicated and the reversing characteristics are not so good. With the synchronous motor, however, both the generator and motor fields must be

excited by an independently driven D. C. generator. (See Fig. 68.)

The induction motor with power factor correction has been also proposed. It has the additional complication of the phase advancer but is claimed to be lighter and less expensive than the

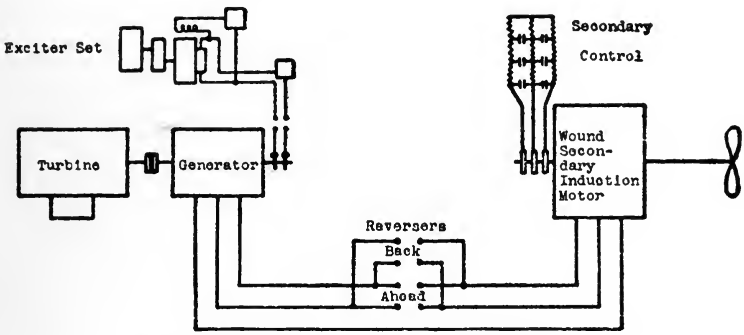



FIG. 70.—Diagram of induction motor installation.

ordinary induction motor and to offer all the advantages of both the induction and synchronous motors.

The motors for ship use are built smaller than those used on shore and consequently are provided with independently driven fans for cooling. (See Fig. 106.)

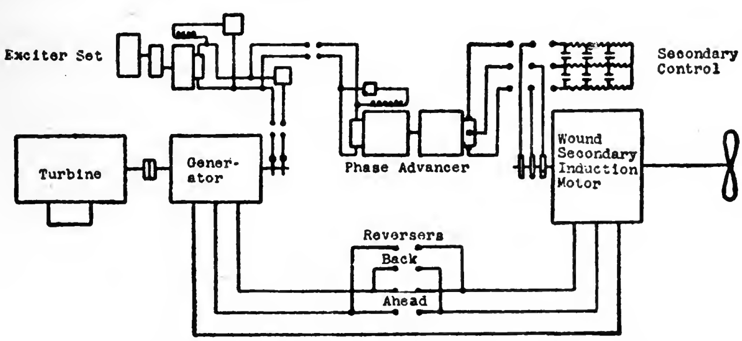



FIG. 71.—Diagram of installation with phase advancer.

**122. Generators.**—A high speed, three phase, two pole A. C. generator is used for supplying the current for the motors. The speed of the generator is fixed by the required r.p.m. of the propeller and the number of poles used in the motor. The larger

the number of motor poles used, the greater will be the speed reduction. Motors for cargo ships have been built with 60 poles and running at 100 r.p.m.; this allows 3,000 r.p.m. for a two pole generator. The upper limit of the number of poles in the motor is probably in the neighborhood of 72, which will allow a maximum generator speed of 3,240 r.p.m. with 90 r.p.m. at the propeller.

All merchant ships so far fitted with electric propulsion have used only one generator and one motor while the battleships have been fitted with two generators and four motors.[2]

The power developed by the motor depends upon the field excitation of the generator and the speed of the turbine. To meet momentary variations in power during maneuvering, turning of the ship and variations in sea conditions, and to provide a high torque for braking and quick reversing and starting, the generator field is over excited.

On the "New Mexico" booster generators are used to add to or subtract from the field excitation without changing the voltage of the exciter generator.

**123. Speed Control.**—On merchant vessels the speed of the propeller, and hence the speed of the ship, is varied by changing speed of the turbine. Varying the speed of the turbine driven generator, varies the frequency of the current and hence changes the revolutions of the motor.

When the speed of the ship is controlled by changing the speed of the turbine, the economy falls off as the turbine r.p.m. are reduced, and the electric drive has no better economy at low speeds than the geared turbine.

In naval vessels two generators are used and the motors are wound so that two combinations of poles can be used. This allows the turbines to run at full speed for several speeds of the propelling motor and thus high steam economy at reduced speeds of the ship is possible.

As merchant ships run practically the entire time at full speed the features used in naval vessels for providing economy at reduced speeds are not required. A high speed passenger ship, however, which has a sea speed of between 20 and 23 knots if fitted with electric drive, should be equipped with two generators or pole changing motors so that high economy can be had at speeds around 16 to 18 knots. This will allow such

[2] See article on generators in *General Electric Review*, Feb., 1921.

a ship to run at reduced speeds with low steam consumption when traveling conditions do not warrant full speed on the passage. For the above conditions the electric drive offers advantages not possessed by the geared turbine.

**124. Starting and Reversing.**—In starting a ship fitted with an induction motor, the turbine is first brought up to idling speed; the reversing switch is put in the "ahead" position; and the generator field switch is closed and excitation established. As soon as the motor comes up to a steady speed the secondary winding is short circuited and the motor is brought up to the desired speed by increasing the speed of the turbine. All operations are performed by levers at one control station.

In reversing when underway at full speed, the turbine speed is reduced to idling speed (about one-third full speed), the field switch is opened and the reversing switch is thrown from "ahead" to "astern" position. The generator field is then excited and as soon as the motor speed has become steady it is brought up to the desired speed as before in step with the turbine. A complete system of interlocks is provided so that no operation can be performed out of its sequence by an inexperienced engineer.

The propelling motor must not only develop a driving torque quickly when reversed but it must also have a braking effect in order to bring the propeller to rest before it is reversed. Motors with special winding are necessary to give these characteristics.

**125. Special Features of the U. S. S. "New Mexico" Installation.**—The electrical equipment on the "New Mexico" consists of two turbine generator units of 10,500 kw. each, running at 2,100 r.p.m.; and four double squirrel cage, pole changing motors running at 167 r.p.m. The motors are direct connected to the propeller shafts. Fig. 107 shows a layout of this installation.

The generators are so wound that two voltage connections are available—a 3,000 voltage and a 4,240 voltage connection. The motor stator windings are wound so that either a 24 pole or 36 pole arrangement is possible.

Between 15 and 17 knots one generator drives four motors and the low voltage generator connection is used; above 17 knots one generator drives two motors and the high voltage connection is used. This arrangement, together with the pole changing motor, gives high efficiency over a wide range of speeds. The speed of the ship is controlled by the speed of the turbines, the

number of motor poles, and the number of generators in operation.

Although the double squirrel cage winding gives a larger motor, it has the advantage of a high reversing torque and a low resistance path at full speed.

For speeds up to 15 knots one generator is used with the low voltage winding and the motors are connected up so that 36 poles are used. The propeller speed, and hence the speed of the ship, is changed by varying the turbine speed. From 15 to 17 knots one generator is used with the low voltage windings as before, but the motors are connected up so that the 24 pole arrangement is used. Variations in the motor r.p.m. over this range is controlled by the speed of the turbine generator.

From 17 knots to full speed both generators are used with the high voltage winding and the motors are connected up for the 24 pole arrangement.

The above arrangement allows the turbines to run at full r.p.m. and maximum efficiency for a number of different speeds of the ship. The variation in the turbine speed is small for large changes in the ship's speed, resulting in a rather flat curve for steam consumption per S.H.P. when plotted against the speed of the ship.

**126. Diesel-Electric Drive.**—The Diesel-electric drive is in principle similar to the turbine-electric drive. Generators driven by medium speed Diesel engines furnish the power for the low speed motor on the propeller shaft. The advantages claimed for this type of propulsion are the same as those of the turbine-electric drive, namely, flexibility of control, economy at low speeds and ease of reversing.

For large merchant ships, the Diesel-electric drive is between 15 and 25 per cent lighter, and probably somewhat cheaper than direct Diesel propulsion, due to the higher speeds used with the Diesel engines. The spaces occupied by the two types are practically the same. The Diesel-electric system, however, is more complicated because of the larger number of Diesel units used and the added electrical equipment. The fuel per S.H.P. to the propeller is about 25 per cent greater due to the electrical transmission losses and the higher fuel consumption of the high speed Diesel engines. No better propeller efficiency can be claimed for the single screw drive at 90 r.p.m. than for the twin screw drive at 115 to 120 r.p.m.

Several small installations of Diesel-electric drive are in operation and some larger ones are now building. Notable among the existing installations are the "Tynemouth," a small cargo ship of 500 s.h.p.; the "Mariner," a trawler of 400 s.h.p.; and the "Guinevere," a schooner yacht of 550 s.h.p. The Western Union cable ship now under construction will have an installation of 2,000 s.h.p.

**127. Description of the Diesel-Electric Drive.**—An installation of this type of machinery on a single screw cargo ship will consist of 4 to 8 Diesel engine generator units running at between 200 and 300 r.p.m. and one double-armature motor running at 90 or 100 r.p.m.

Diesel engines of small powers and 200 to 250 r.p.m. can now be obtained from a number of builders and because of the higher r.p.m., and in some cases standardization, sell at a lower cost per horsepower than engines of large power.

Because of the large number of generators used and the difficulty of close regulation with Diesel engines, direct current has been adopted in place of the alternating current used with turbo-electric drive. The generators are operated in series, with generator voltage control which gives a simple and very efficient arrangement.

The installation on the Western Union cable ship consists of four 410-kw. 250-volt Diesel generating sets running at 250 r.p.m. and two 1,000 s.h.p. motors turning 120 r.p.m. Two generators connected in series supply current at 500 volts to one motor. When a single screw arrangement is used a double-armature motor is adopted with 500 to 750 volts on each armature.

The speed of the motor is controlled by varying the excitation of the generator field. The motor fields are excited from a separate source which is kept constant and in one direction at all speeds. The speed on the motor will thus vary directly with the generator voltage and the propeller speed can be controlled directly from the field rheostat. Full speed ahead or astern can be obtained at the motor without opening any circuit by simply varying or reversing the generator field current. The control is thus almost ideal and even better than that of the turbo-electric drive.

Advocates of the Diesel-electric drive frequently propose the use of Diesel engines of extremely high r.p.m. to decrease the

size, weight, and cost of the installation. As pointed out in Chap. XIII, Diesel engines running at speeds much over 250 r.p.m. are not feasible today for continuous commercial service. Submarine engines have been built operating at speeds as high as 450 r.p.m., but these r.p.m. are maintained only for short runs at full speed. The present day tendency is to reduce the full speed r.p.m. and the latest types of submarine engines run at 350 to 375 r.p.m. at full speed.

With the Diesel-electric drive, reversing gear can be eliminated on all the engines and air starting gear can be omitted from all but one engine, the remaining sets being started by using the generators as motors. However, a large merchant ship installation will be extremely complicated with 6-cylinder engines and the large number of valves to keep in order. While this type of machinery apparently offers no advantages for large merchant ships it has many valuable features for small craft and vessels of special types.

TABLE XI.—REPRESENTATIVE ELECTRICALLY PROPELLED SHIPS

| | Current | No. generators | No. propellers | Motor poles | Gen. r.p.m. | Motor r.p.m. | Power factor | Type of motor | Voltage | S.h.p. 1 motor | Frequency | Speed control |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Jupiter..... | AC | 1 | 2 | 36 | 1,990 | 110 | .... | Induction | 2,300 | 2,750 | 33.2 | Turbine |
| New Mexico. | AC | 2 | 4 | 24<br>36 | 2,100 | 167 | .78 | Double squirrel cage | 3,000<br>4,242 | 7,250 | 35 | Turbine & poles |
| Eclipse..... | AC | 1 | 1 | 60 | 3,000 | 100 | .70 | Induction | 2,300 | 3,000 | 50 | Turbine |
| Tampa..... | AC | 1 | 1 | 46 | 3,000 | 130 | 1.00 | Synchronous | 2,300 | 2,600 | 50 | Turbine |
| Biyo Maru[2] | AC | 2[2] | 1 | .... | 3,600 | 600 | .... | Induction | 1,000 | 1,360[2] | 60 | .......... |
| Cuba....... | AC | 1 | 1 | .... | 3,000 | 100 | 1.00 | Synchronous | 1,100 | 3,000 | 50 | Turbine |
| Mariner[1].... | DC | 2 | 1 | .... | 350 | 200 | .... | DC | 250 | 400 | ... | Engine & motor |
| Cable ship[1].. | DC | 4 | 2 | .... | 250 | 120 | .... | Double. arm. | 500 | 1,000 | .... | Rheostat |
| Guinevere[1] | DC | 2 | 1 | .... | 225 | 220 | .... | DC | 250 | 550 | .... | .......... |
| Fordonian[1].. | DC | 2 | 1 | .... | 200 | 120 | .... | Double arm. | 250 | 850 | .... | .......... |

[1] Diesel electric.

[2] Ljunstrom radial flow turbines 225 lbs. and 250° superheat. 2 motors of 1,360 hp. each geared to one propeller. Motors run at 600 r.p.m., propeller at 80 r.p.m. (See Fig. 112.)

# CHAPTER XIII

## THE DIESEL ENGINE

**128. The Diesel Cycle.**—A $PV$ diagram of the Diesel cycle is shown in Fig. 72 and a $T\phi$ diagram in Fig. 73. The heat is added with an increase in temperature along the constant pressure line $ab$; adiabatic expansion takes place along $bc$; the heat

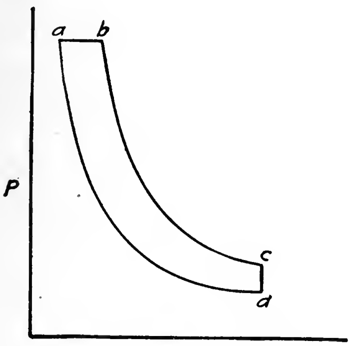



FIG. 72.—PV diagram of Diesel cycle.

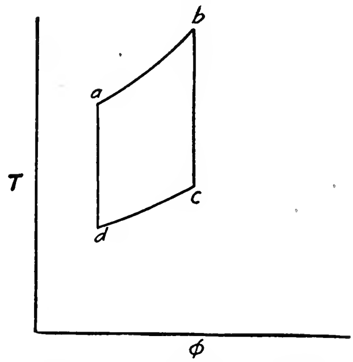



FIG. 73.—T$\phi$ diagram of Diesel cycle.

is rejected along the constant volume line $cd$; and an adiabatic compression from $d$ to $a$ completes the cycle.

The heat received along $ab$ is,

$$Q_1 = c_p(t_b - t_a)w$$

The heat rejected along $cd$ is,

$$Q_2 = c_v(t_c - t_d)w$$

The efficiency of the cycle becomes,

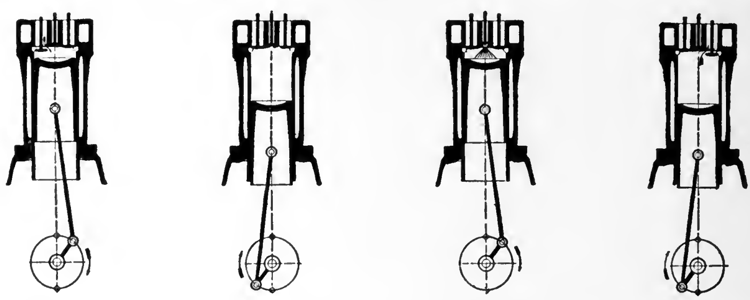
$$\frac{Q_1 - Q_2}{Q_1} = \frac{c_p(t_b - t_a) - c_v(t_c - t_d)}{c_p(t_b - t_a)} = 1 - \left(\frac{1}{n}\,\frac{(t_c - t_d)}{(t_b - t_a)}\right)$$

where
$$\frac{c_p}{c_v} = n.$$

This can be reduced to,

$$\text{Efficiency} = 1 - \left\{ \frac{1}{n}\left(\frac{V_a}{V_d}\right)^{n-1} \times \frac{\left(\frac{V_b}{V_a}\right)^n - 1}{\left(\frac{V_b}{V_a}\right) - 1} \right\}$$

The expression for efficiency involves the term $\left(\frac{V_a}{V_d}\right)^{n-1}$ as in the Otto cycle, showing that the efficiency depends upon the range of compression. It also involves the ratio of cut-off volume to clearance volume, $\left(\frac{V_b}{V_a}\right)$. The efficiency of the Diesel engine is higher than that of engines working on the Otto cycle because of the higher compression pressure used. Theoretically



(a) First stroke (admission stroke) Piston travels down; admission valve open; cylinder is being filled with pure air

(b) Second stroke (compression stroke). Piston travels up; all valves closed; air in cylinder is being compressed.

(c) Third stroke (power stroke). Piston travels down; fuel valve open at top dead-center, but closed at fraction of stroke; gases expand.

(d) Fourth stroke (exhaust stroke). Piston travels up; exhaust valve open; burnt gases are being expelled from cylinder.

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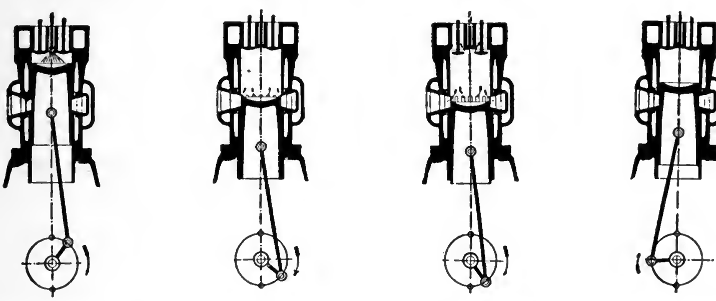
FIG. 74.—Four-cycle Diesel engine.

the shorter the cut-off the higher the efficiency; but this does not exist in practice.

The Diesel engine is named after the inventor of the cycle, Dr. Rudolph Diesel. In working up his engine, it was the hope of Diesel to obtain an isothermal admission instead of a constant pressure admission, and thus approach the Carnot cycle.

**129. Mechanical Operation.**—The Diesel engine operates on both the two and four stroke cycle, the same as the Otto engine. Heavy grades of oil are used for fuel and no electric sparking device is needed as the fuel is ignited by the heat of compression.

In the four stroke cycle, or, as it is commonly called, the four cycle engine, four strokes of the piston are required to complete the cycle. Consider view (*a*) in Fig. 74. As the piston moves down, the admission valve is opened mechanically and a charge of pure air is drawn in. At the end of the down stroke the admission valve is closed and the piston begins its upward stroke (*b*), compressing the air to about 500 lbs. per square inch. When the piston reaches the end of this upward stroke, the spray valve opens for a small fraction of the stroke and finely atomized fuel oil is sprayed into the combustion space. Due to the heat of compression the fuel is ignited and burns during the time that the fuel valve is open. The fuel spray valve then closes and the gases expand, forcing the piston down on



(a) Piston traveling down; all valves and ports closed; fuel injected and burnt during first part of stroke; gases expand.

(b) Piston traveling down; piston uncovers exhaust ports; burnt gases escape through exhaust ports, reducing their pressure to atmospheric

(c) Piston traveling down; exhaust ports still uncovered; scavenging valves open, air under light pressure enters through scavenging valves, and blows gases out of cylinder.

(d) Piston traveling up; piston covers exhaust ports; scavenging valve closed; air in cylinder is being compressed.

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Fig. 75.—Two-cycle Diesel engine.

its working stroke (*c*). At the end of the third stroke the exhaust valve in the top of the cylinder opens and the piston rises on its fourth stroke (*d*), forcing the burnt gases out. The exhaust valve then closes and the cycle is repeated. It will be observed that four strokes are necessary to complete the cycle and, as the engine is only single acting, there is only one working stroke during the cycle.

The two stroke cycle or two cycle engine requires but two strokes to complete the cycle. In Fig. 75 at (*a*) the piston is at the top of its stroke with pure air compressed in the clearance

space. The fuel valve opens and a small quantity of finely atomized oil is sprayed in. The fuel valve then closes and the piston starts down on its working stroke, the gases expanding behind the piston. As the piston reaches the end of its stroke (*b*) it uncovers ports in the cylinder walls and the burnt gases escape. An instant later (*c*) the scavenging valve in the top of the cylinder is opened mechanically and air under low pressure is forced in to drive out the burnt gases and fill the cylinder with pure air. The piston then starts up on its second stroke (*d*) and compresses the air. At the top of the stroke the spray valve again opens and the cycle is repeated.

It will be observed that only two strokes are required to complete the cycle and that there is a working stroke once every revolution instead of every other revolution as is the case with the four cycle engine.

In the two cycle engine the scavenging of the cylinder and the filling of the cylinder volume with air take place at the end of the down stroke by using air under low pressure, while in the four cycle engine two idle strokes are used, one for scavenging and one for drawing in the fresh air charge. The two cycle engine requires a scavenging air pump to supply the scavenging air at the end of the working stroke.

Practically all marine Diesel engines are single acting. The notable exception is the double acting engines of the motorship "Fritz" which are three cylinder, 850 S.H.P. engines, 17 3/4 × 28 in. running at 120 r.p.m.

**130. Fuel Injection Systems.**—There are two methods of injecting the fuel oil into the cylinder of the Diesel engine—air injection and solid injection. Air injection is the most generally used and is the one first used with Diesel engines.

The fuel system consists of fuel pumps, fuel oil piping, fuel spray valve, high pressure air compressor, air coolers, spray air flasks and air piping and control gear at the control station.

The fuel or spray valve is one of the most important parts of the Diesel engine and each manufacturer has his own particular type. The fuel valve has a dual function: (1) that of a valve for admitting the fuel supply to the cylinder at the proper point in the cycle; and (2) that of an atomizer for breaking the fuel up into a fine fog like spray.

**131. Air Injection System.**—A measured amount of fuel oil

is delivered to a pocket in the fuel valve by a plunger of the fuel pump while the fuel valve is closed. At the proper point in the cycle the fuel valve is opened by the valve gear and the fuel is forced into the combustion space by high pressure air. There is no valve in the h.p. air line and the fuel valves are always in communication with the spray air flasks. The object of supplying air is to atomize the fuel in the passages of the fuel valve and to insure that it enters the cylinder in a finely divided state so that complete and rapid combustion will take place.

The amount of fuel delivered to the spray valve depends upon the length of the working stroke of the fuel pump. One method of controlling the fuel supply is to hold open the suction valves of the pump during part of the pressure stroke and thus allow part of the fuel to be returned back to the supply tank. Both the hand and governor control are attached to the suction valves of the pump and thus the control of the fuel supply, and hence the speed of the engine, are regulated as desired. Separate plungers are provided in the fuel pump for each cylinder, or each pair of cylinders that can be grouped, as the fuel supply must be delivered to the fuel valve while it is closed. The fuel pump is always driven by the main engine.

The spray air is compressed in a three stage compressor to a pressure between 800 and 1,200 lbs. per sq. in. This compressor is driven from the main crankshaft and is built as an integral part of the engine. Because of the heat resulting from the high compression, air coolers are needed between the stages for cooling the air. Salt water is used for cooling. Heavy high-pressure air piping is fitted from the compressors to the coolers and from the last cooler to the spray air flasks and thence to the fuel valves.

**132. Solid Injection System.**—In this system the fuel is forced into the cylinders under an extremely high pressure of between 4,000 and 9,000 lbs. per sq. in. Atomization is accomplished both by the design of the spray valves and the high pressure used. The solid injection system is much simpler than the air injection system as the high pressure air compressors, air piping, and spray flasks are eliminated. On the other hand, the fuel piping, fuel valve, and pump must be built to withstand a very high pressure. The fuel oil pump is one of the most delicate and important parts of the Diesel engine for it must

deliver the correct amount of fuel to each fuel valve. The speed and regularity of the engine depend directly upon the amount of fuel supplied to the valve. The supply used per cycle by each valve is extremely small and leakage at the pump or in the fuel piping cannot be tolerated. It is evident that the design of such a pump working under these high pressures requires special attention.

**133. Comparison of Air and Solid Injection.**—The engine using the solid injection system is simpler, lighter, occupies less space, and has a higher mechanical efficiency due to the elimination of the engine driven h.p. air compressor. The engine also can be run at lower r.p.m. due to the absence of the cold air blast inherent with the engine using the air injection system.

One of the greatest advantages of the solid injection system is the absence of the injection air blast which cools the combustion space. In many makes of Diesel engines the amount of injection air and the duration of injection is constant, regardless of the speed of the engine or the amount of fuel used per stroke. The large and excessive supply of cold air at low r.p.m.'s, when the amount of fuel per stroke has been cut down, makes steady running at low speeds difficult and often impossible with the air injection system.

The heavier and more viscous oils can be better used with the solid injection system as they require more heat and higher pressures to break them up into a fine spray. The objections to the solid injection system are, the high oil pressures used, the greater difficulty in keeping the pump and oil piping tight, and the higher fuel consumption due to poorer atomization. The makers of engines using solid injection deny that the fuel consumption is any higher. Recent trials seem to indicate that with increased experience with this system, better atomization has been brought about and the fuel consumption is as low as that obtained with air injection. If this is borne out in practice, the solid injection system has most of the advantages in its favor.

**134. Diesel Engine Auxiliaries.**—A marine Diesel engine is supplied with the following equipment and auxiliary systems:

1. Fuel oil system.
2. Spray air system.
3. Lubricating oil system.
4. Circulating (cooling) water system.

5. Piston cooling system.
6. Air starting system with auxiliary air compressor.
7. Valve gears.
8. Reversing and speed control gear.
9. Auxiliary generating sets (to supply motor driven auxiliaries and winches).
10. Exhaust system.
11. Scavenging air system.

The second system, as pointed out in the previous article, is not used with solid injection and the last system is only used with two cycle engines.

**135. Lubricating Oil System.**—The lubricating oil system is used to supply lubricating oil to the shaft bearings, connecting rod bearings, pistons, camshafts and gears, cross head guides, and thrust bearing. Lubricating oil pumps are fitted in duplicate and are either driven by the main engine or independently by an electric motor.

Two systems are ordinarily installed. One system, supplied by the main lubricating oil pump, delivers oil to the main bearing, thence to the crankpin bearings and finally up the connecting rod to the wrist pin or to the cross head pin and guides. The oil then drains by gravity to the crankcase and thence to the sump tank. The amount of oil circulated through the system is between 1 and 2 gallons per B.H.P. An oil cooler is fitted in the circuit between the engine and the pump to cool the lubricating oil. The engine is fitted with a tight housing to keep the oil from being splashed about the engine room. A secondary system operated by a mechanical oiler supplies oil to the main and air compressor pistons.

**136. Circulating Water System.**—Circulating water must be supplied to the cylinder jackets, the cylinder heads and to the water jackets of the exhaust manifold and piping. In large engines, the exhaust valves and main bearings are also water cooled. The circulating pump is practically always independently driven by an electric motor. Salt water is almost invariably used for cooling although the Doxford two cycle engine uses fresh water which is recirculated and cooled by a separate salt water circulating system. A fresh water cooling system adds materially to the weight and space occupied by the machinery; yet for vessels operating in shallow and muddy water it

may be a necessity in order to keep mud and sediment from filling up the cylinder jackets.

When salt water is used for cooling it is necessary to keep the circulating water system in operation after the engine has been shut down so that salt scale will not be deposited in the jacket space.

The practice of running both the circulating and lubricating oil pumps after stopping should always be practiced on large engines to keep the lubricating oil from hardening and thus clogging up the passages in the engine.

**137. Piston Cooling System.**—Because of the excessive amount of heat generated in large engines, and in all two cycle engines, piston cooling has to be resorted to. Either lubricating oil or fresh water is used for this purpose. Salt water cannot be used because of the high temperatures and the resultant formation of scale that would result.[1] In small engines the main lubricating supply is used. The oil enters the hollow piston head after leaving the wrist pin and drains from the piston to the crankcase by gravity. In large engines either a separate supply of oil or water is used for piston cooling. This necessitates elaborate linkages or sliding joints in the piping because of the reciprocating motion of the piston. Coolers must also be fitted in the system and a separate motor driven pump supplied.

**138. Air Starting.**—All marine engines use compressed air for starting, reversing, and maneuvering. Air under 400 to 500 lbs. pressure is used for this and is stored in large tanks in the engine room (see Fig. 80). The compressed air supply is furnished either by the engine driven compressor (when air injection is used) or by a separate compressor driven by an auxiliary Diesel engine (Fig. 80).

**139. Engine Control and Reversing.**—Diesel engines are started by compressed air and as soon as the engine comes up to speed the air is cut off from the starting valves and the fuel pump is brought into action. In reversing, the fuel supply is shut off, the valve gear shifted and the engine started in the reverse direction by compressed air. When the engine is up to

[1]The motorships "William Penn" "Californian" and "Missourian" which are fitted with Burmeister and Wain engines use fresh water for cooling the cylinder jackets and heads and salt water for cooling the pistons, bearings, and exhaust piping.

the proper speed the air supply is cut off and the fuel valves brought into action the same as when going ahead.

Reversing is generally accomplished in Diesel engines by bringing a second cam into operation by bodily shifting the camshaft.

The method used in reversing the Sulzer engine is shown in Fig. 76. Two cams are fitted on the camshaft for each fuel valve. The fuel valve is actuated by these cams through a lever as shown in Fig. 76. In going ahead the cam ($a$) operates the valve through roller ($a_1$). When it is desired to reverse the roller ($a_1$) is lifted clear of the cam ($a$) and the roller ($b_1$) is brought into operation with cam ($b$). At the same time that the rollers are shifted the air starting valves are opened and

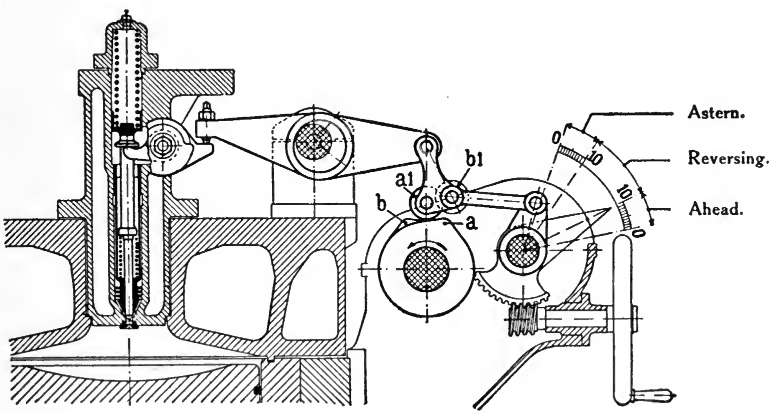



Fig. 76.—Reversing gear of Sulzer engine.

compressed air is admitted to the cylinders to start the engine astern.

The handwheel shown in Fig. 76 also regulates the opening of the fuel spray valve by varying the clearance between the roller and cam. Thus, in this engine, the speed control is obtained by the amount of opening of the fuel valve instead of having a constant valve lift and controlling the speed by the stroke of the fuel pump alone as is customary in most engines.

As already pointed out, in many engines the lift of the fuel valve, and hence the supply of air admitted to the combustion space, is constant for all speeds. The fuel supply, however, is controlled by the stroke of the fuel pump which is shortened as the speed and load are reduced. The Sulzer engine, described

above, is one of the exceptions to this. In some engines there is a control to the suction valve of the air compressor which allows the pressure of the spray air flasks to be reduced at light loads.

**140. Scavenging System.**—Scavenging air is supplied for all two cycle engines as pointed out in Art. **129.** This scavenging air may be supplied by valves in the head as in Fig. 75 or by port scavenging (Figs. 77 and 78). The latter is the latest and best method as the head casting is not complicated by the additional valve and the driving gear is simpler and quieter. The scavenging air is at a low pressure and is supplied by an engine driven scavenging pump. Sulzer Bros. (Art. **143**) have replaced their engine driven scavenging pump by a motor driven blower (see Fig. 80), thus increasing the mechanical efficiency and shortening the length of the engine.

**141. Mechanical Efficiency.**—The mechanical efficiency of a Diesel engine varies between 60 and 80 per cent and is much lower than any of the other prime movers, due to the large number of engine driven auxiliaries. The mechanical efficiency can be improved by using independently driven auxiliaries, but in such a case care must be exercised that the fuel consumption of all these auxiliaries is charged up to the main engines. The fuel per B.H.P. should be the fuel used by the main engine and all main engine auxiliaries divided by the *B.H.P. of the main engine* and *not* by the combined B.H.P. of main engine and auxiliaries. The unit of power for the Diesel engine is the brake horsepower (B.H.P.) which is equivalent to the S.H.P. The indicates horsepower (I.H.P.) is frequently stated but is of little value because indicator cards can seldom be taken and are not reliable, due to the high temperature effects. Fuel consumptions based on I.H.P. are frequently stated but are of no value. They are, in fact, often misleading. The horsepower delivered to the shaft is the true criterion of power for all propelling units. Indicator cards are frequently taken on Diesel engines as a continuous plot in order to study the behavior of the spray valve.

**142. Engine Details.**—The Diesel engine has rightly been called "an engine of details." Far more troubles have arisen over small details than from errors and inexperience in general design. The high pressure and high temperatures require careful attention to materials and quality of castings. Cylinder

heads with their many valve openings and complicated jacket cores have been a constant source of trouble. Bearing pressures, salt water in the lubricating oil supply, torsional vibrations of the shafts, exhaust valves, and fuel valves causing improper

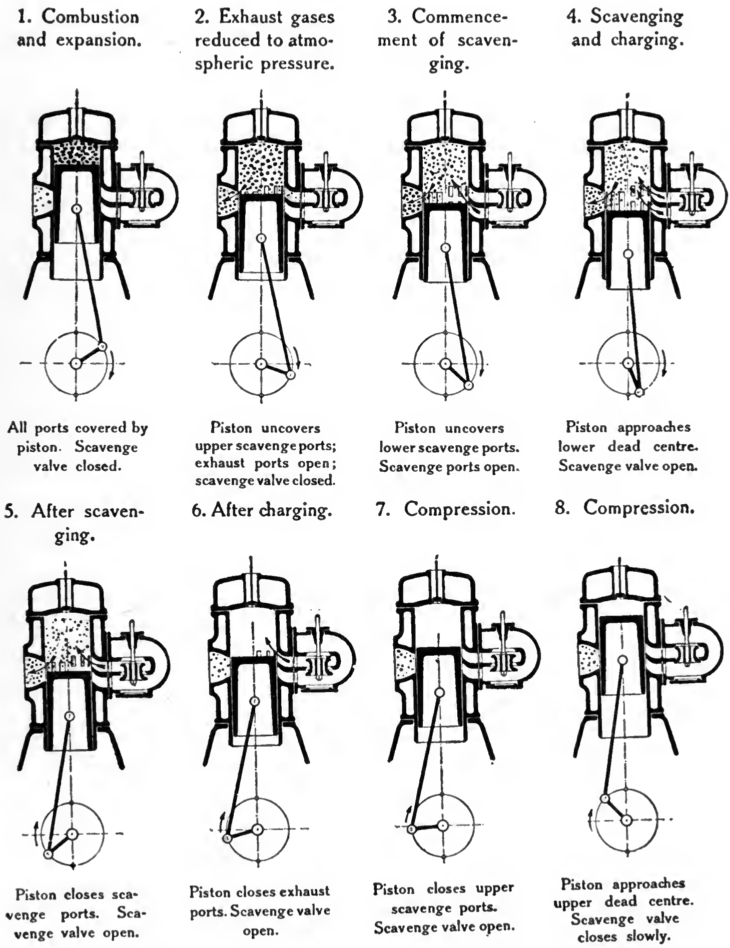



FIG. 77.—Sulzer two-cycle Diesel engine with port scavenging.

atomization and pre-ignition, and cracked pistons are some of the difficulties that have confronted the engineer in the development of the Diesel engine.

The difficulties with details have increased as the r.p.m. of

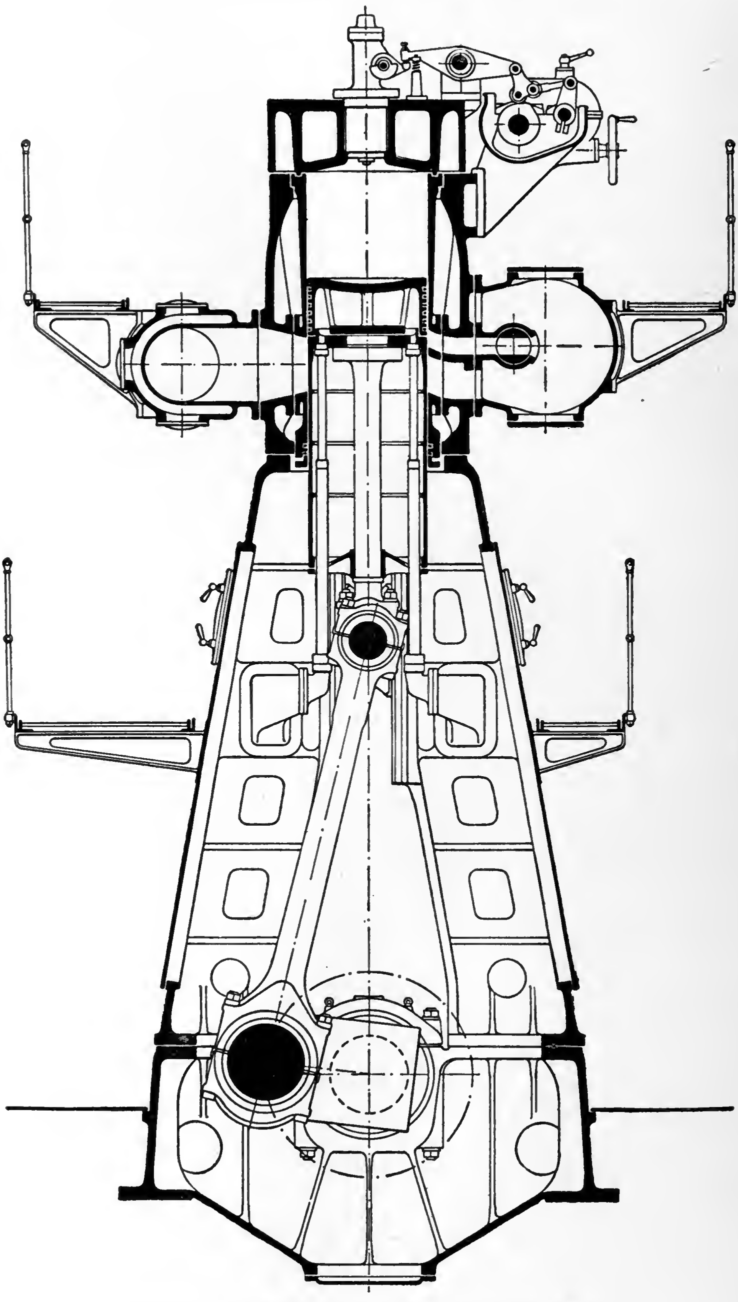


Fig. 78.—Cross section of Sulzer two-cycle engine.

the engines have been increased. Revolutions over 250 are not recommended for commercial work and revolutions of 375, while used in submarine engines, are only maintained for short runs.

**143. The Sulzer Two-Cycle Engine.**—This engine, built by by Sulzer Brothers of Winterthur, Switzerland, is one of the most reliable two-cycle engines built and has many valuable and novel features in its design. The Sulzer engine is of the port scavenging type. A diagram of the engine cycle is shown in Fig. 77. Two scavenging ports are provided—a lower one which is uncovered by the piston in its downward stroke and an upper port fitted with a valve (Fig. 77 shows a poppet type scavenging valve but the latest practice is to use a rotary valve as shown in Fig. 78). The valve in the upper port does not open until the lower one has been uncovered, but it stays open (Fig. 77) after the exhaust port and lower scavenging port have been closed by the upward movement of the piston. This allows a larger supply of fresh air to be blown into the cylinder than when only one scavenging port is used. Thus the cylinder is filled with a larger supply of air and can develop more power per stroke because of the higher m.e.p. A cross-section of the engine is shown in Fig. 78. The use of the port scavenging system eliminates all the valves in the head except the fuel valve and air starting valve. These are fitted in a common casing at the center of the cylinder head, thus requiring only one opening through the head casting. This is an important feature as it greatly simplifies the jacketed head casting—a constant source of trouble in the development of the Diesel engine. Piston cooling is carried out by the use of telescoping pipes clearly shown in Fig. 78.

In the smaller powers the engine is fitted with a double acting scavenging pump while, for powers above 1,000 S.H.P., electric driven turbo-blowers are used. Figure 79 shows a view of a Sulzer 1,350 S.H.P. engine fitted for using the motor driven blower for scavenging. Figure 80 shows an engine room arrangement for a twin screw ship with two 4,500 S.H.P. Sulzer engines.

Due to the absence of exhaust, inlet and air scavenging valves a two-cycle port scavenging engine, similar to the Sulzer engine, can operate successfully on the inferior and heavy asphaltum oils and Mexican fuel oils as pointed out in Art. **147**.

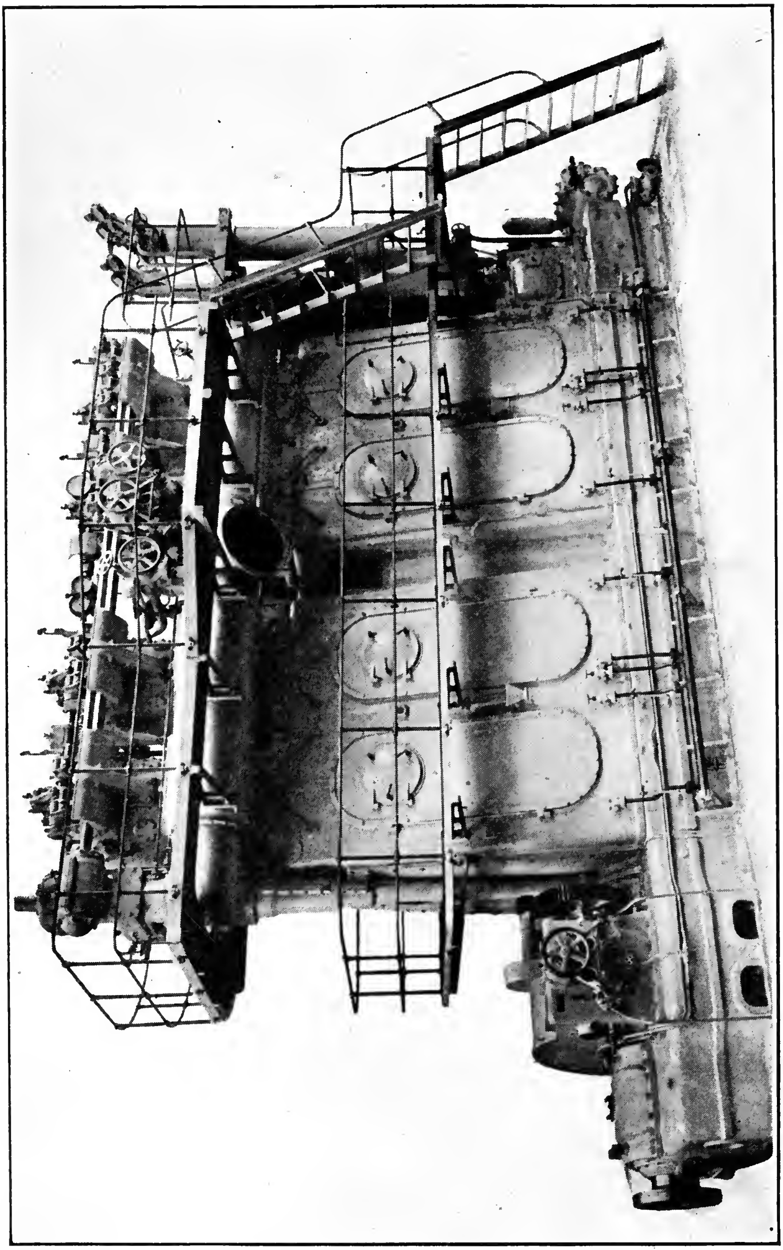


FIG. 79.—Sulzer two-cycle engine fitted for scavenging with motor-driven blower.

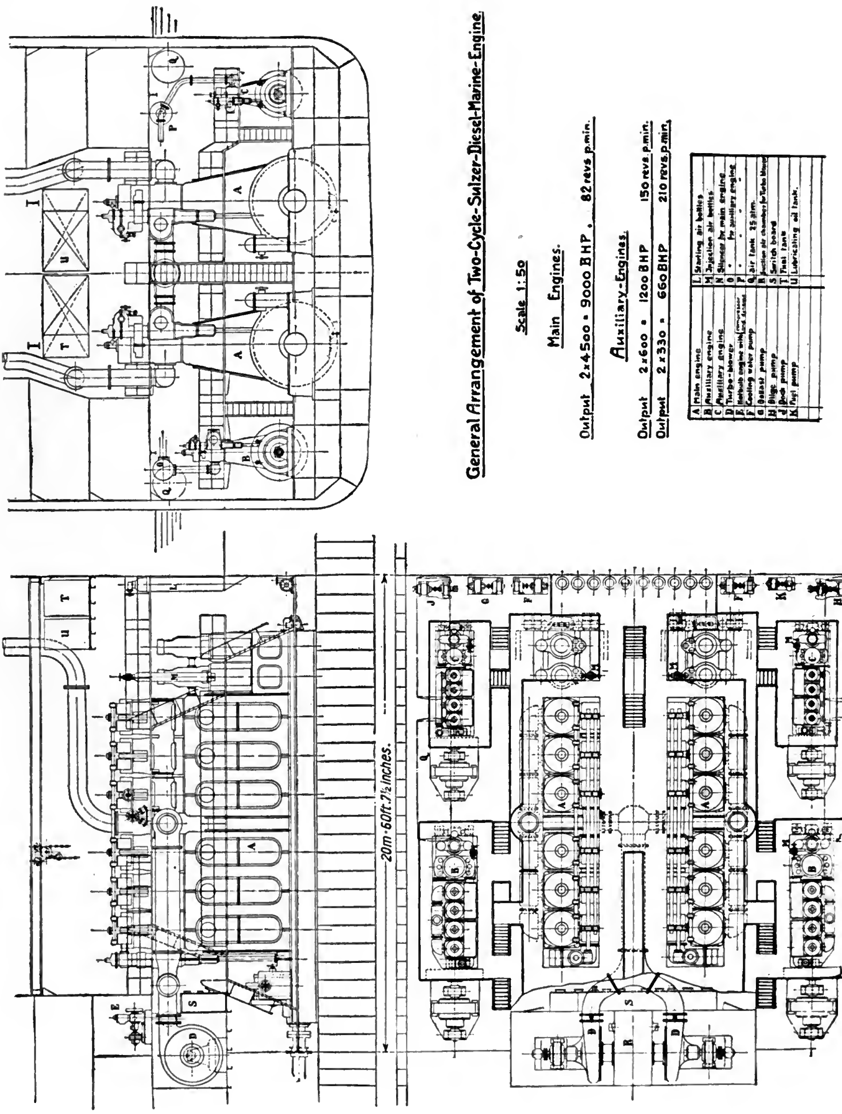

General Arrangement of Two-Cycle-Sulzer-Diesel-Marine-Engine.
Scale 1:50
Main Engines.
Output 2x4500 = 9000 BHP, 82 revs p.min.
Auxiliary-Engines.
Output 2x600 = 1200 BHP 150 revs p.min.
Output 2x330 = 660 BHP 210 revs p.min.
A Main engine
B Auxiliary engine
C Auxiliary engine
D Turbo-blower
F Cooling water pump
G Ballast pump
H Bilge pump
J Deck pump
K Fuel pump
L Starting air bottles
M Injection air bottles
N Silencer for main engine
O " for auxiliary engine
Q Air tank 25 atm.
S Switch board
T Fuel tank
U Lubricating oil tank.
20 m · 60 ft. 7½ inches.

FIG. 80.

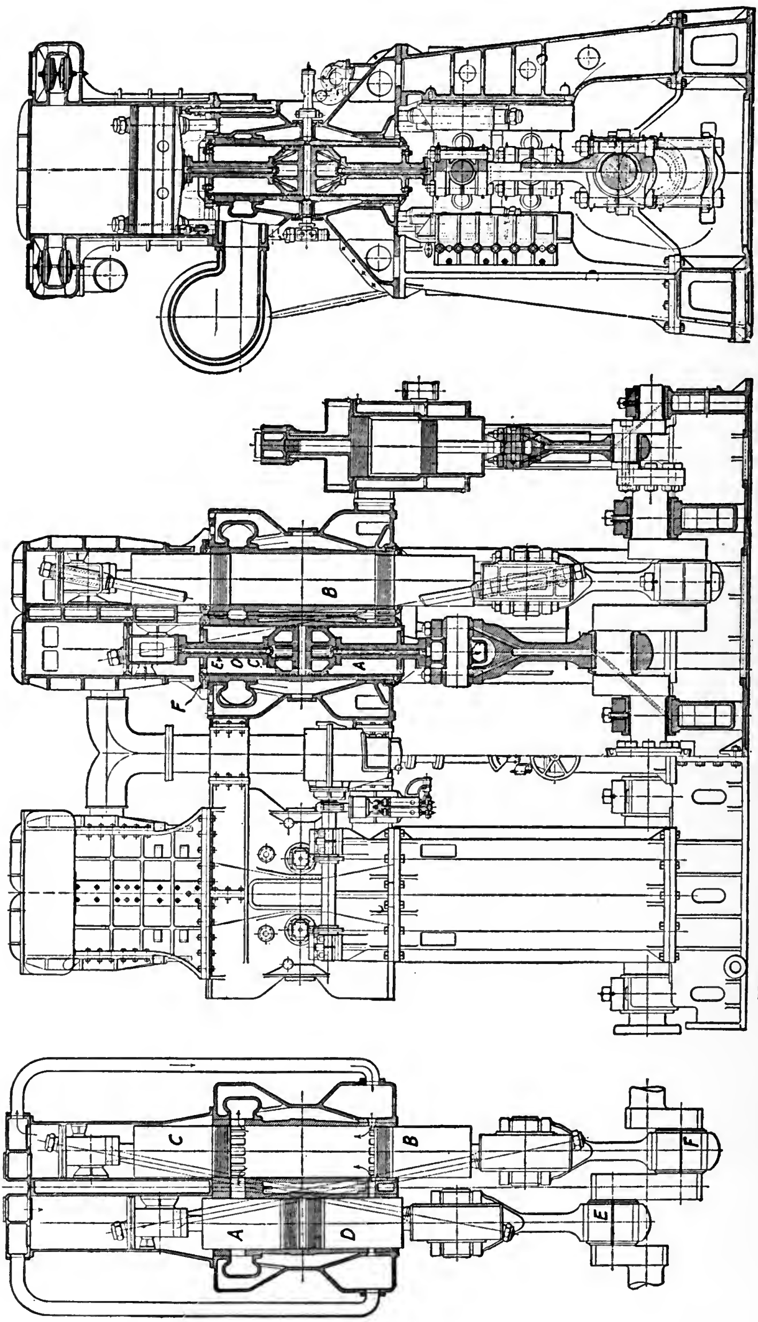



FIG. 81.—Cammellaird-Fullagar opposed-piston Diesel engine.

**144. Cammellaird-Fullagar Engine.**—This engine is of the Junkers two-cycle type using opposed pistons. While most opposed engines require three cranks per cylinder, making a complicated mechanical design, the Cammellaird-Fullagar engine uses only two cranks for each pair of cylinders. Thus the advantages of the opposed piston type are secured without the disadvantages.

As shown in Fig. 81, the cylinders are arranged in pairs. Each cylinder is open at both ends and fitted with two pistons. The upper piston $A$ of one cylinder is connected to the lower piston $B$ of the second cylinder by diagonal tie rods; and the upper piston $C$ of the second cylinder is connected to the lower piston $D$ of the first cylinder in a similar manner. When combustion takes place in the first cylinder the pistons $A$ and $D$ are forced apart, $A$ moving upward and $D$ downward. The pistons in the second cylinder come together on the compression stroke as $A$ and $D$ separate on the working stroke. Thus crank $E$ is forced down and crank $F$, which is at 90° to $E$, is forced up.

The fuel spray valves are placed in the center of the cylinder between the two pistons. The exhaust ports are placed at the top of the cylinder and are uncovered by the upper piston; the scavenging air is admitted through a complete ring of ports located at the bottom of the cylinder which are uncovered by the lower piston. This allows the scavenging air to transverse the entire cylinder space and insures excellent scavenging. The scavenging pumps are rectangular in cross-section and are located on top of the cylinders.

The advantages claimed for this engine, many of which are inherent with the Junkers type, are as follows:

1. Piston loads balanced so that no stress comes on columns or cylinders.
2. Cylinder heads with all their complications eliminated.
3. One-third to one-half the fore and aft space required for four-cycle engine of same power.
4. Good balance due to pistons moving in opposite directions.
5. Rate of expansion of gases double that for other engines of same piston speed, allowing operation at low r.p.m.
6. Excellent scavenging.

**145. Burmeister and Wain Engine.**—This engine is the best known of the four-cycle types, is built by a large number of licensees, and has been used in motorships more widely than

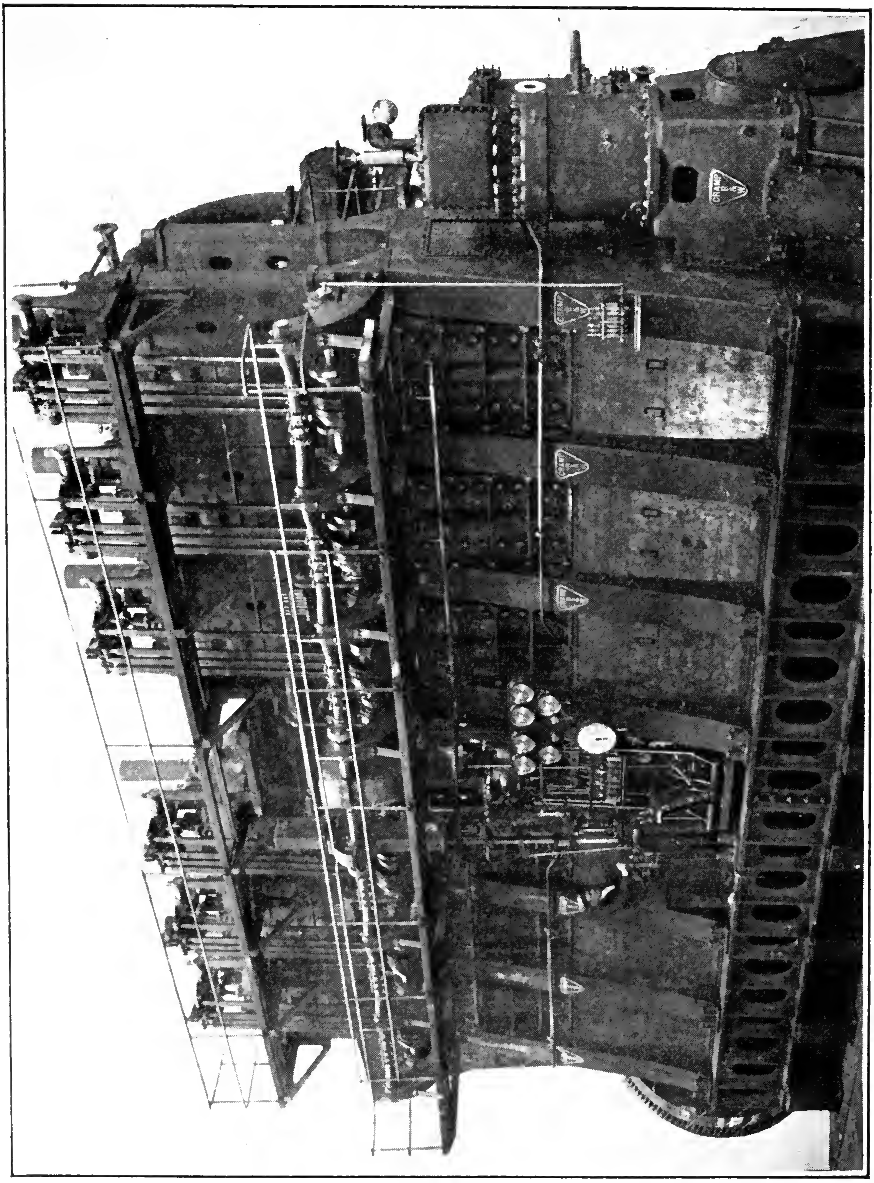


Fig. 82.—Cramp-Burmeister and Wain Diesel engine of 1,750 S.H.P. at 115 r.p.m.

any other make. Figure 82 shows a Burmeister and Wain engine of 1,750 S.H.P. at 115 r.p.m. built by the William Cramp & Sons S. & E. B. Co. of Philadelphia. This engine is of the cross-head type and has six cylinders 29 1/8 in. bore by 45 1/4 in. stroke. Figure 83 is a cross-sectional view through the engine and shows clearly the piston and cross-head construction, piston cooling arrangement and valve gear.

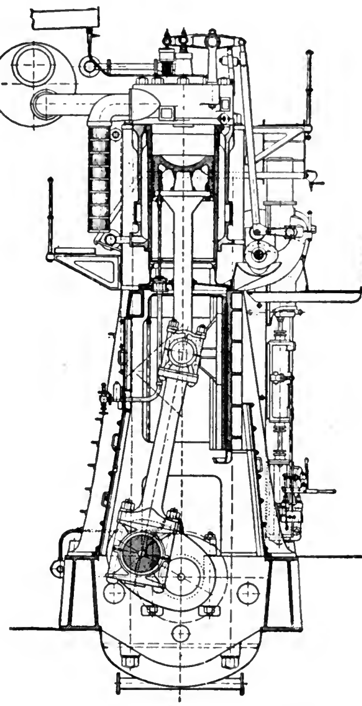


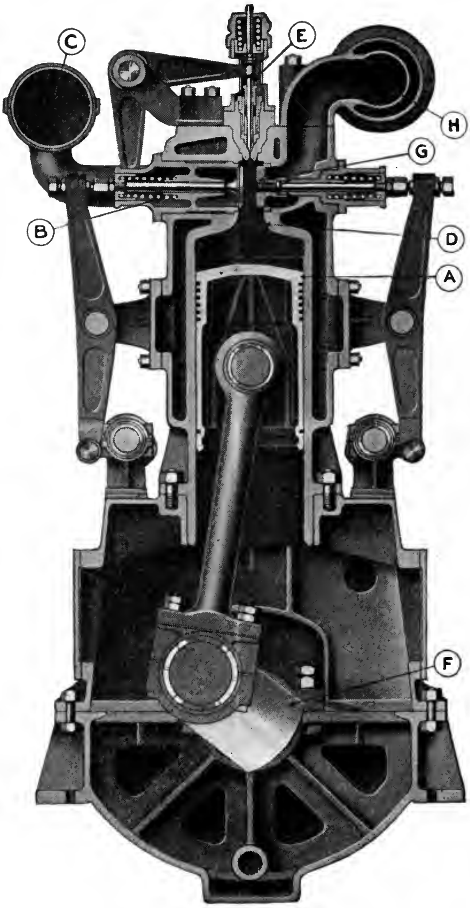
Fig. 83.—Cross-section of B. & W. engine.

Burmeister and Wain have also developed a long stroke engine of 85 r.p.m. for single screws in which the stroke-bore ratio is slightly over 2.0. The standard designs for twin screw ships has a stroke-bore ratio of about 1.5.

**146. Nelseco Engine.**—The New London Ship & Engine Co. has developed a small power type of four-cycle Diesel engine. Standard commercial engines of four, six and eight cylinders are built in horsepowers ranging from 60 to 600 s.h.p. and r.p.m. ranging between 200 and 350. This company also has built a large number of submarine engines of from 700 to 1,000 s.h.p. which run at 350 and 375 r.p.m. Figure 84 shows a section through the Nelseco engine. The inlet, exhaust and spray valves, camshafts and valve gear are clearly shown. This engine is typical of a large number of Diesel engines of small powers and moderately high r.p.m. that are available for the propulsion of small craft and for units in larger ships using Diesel-electric drive.

**147. Present Day Trend and Practice.**—Table XII shows the sizes of Diesel engines used in some of the largest motorships. Nearly all motorships have been built with twin screws because

Diesel engines of large power have not been available and because shipowners have considered the Diesel engine too precarious to install a single engine. With the increase in horsepower of Diesel engines and its greater reliability, the single screw motorship will no doubt predominate.




A. Piston.
B. Inlet valve.
C. Inlet manifold.
D. Cylinder.
E. Fuel valve.
F. Crank shaft.
G. Exhaust valve.
H. Exhaust manifold.

FIG. 84.—Cross-section of Nelseco Diesel engine.

Diesel engines of 2,700 S.H.P. are already in successful operation and larger sizes are already designed and firms are ready to undertake their construction. The twin screw, 16 to 17 knot motor liner of 12,000 S.H.P. will probably appear in the near future.

Diesel engines of high r.p.m. and double acting engines so far have not been successful but the double acting engine will no doubt be developed.

Today the four-cycle engine predominates; but because of its lighter weight and smaller fore and aft length the two-cycle engine is bound to be used more and more, especially in ships of moderately high speeds.

The one great drawback to the Diesel engine today is that it requires a lighter and more expensive grade of fuel oil. Prices of Diesel engine oil vary from 25 to 100 per cent more than steamship "fuel oil." When the cost is twice that of fuel oil the advantages of cost, space, and economy are all in favor of the geared turbine ship.

Heavy tar oils have been successfully used with Diesel engines on shore by heating the oil, but such a system is not feasible on shipboard. The two-cycle engine, especially the port scavenging type, is better adapted for using the heavier fuel oils because of the absence of valves.

The great necessity in future development with Diesel engines is to make them available for burning the heavier Mexican and Texas fuel oils and residual oils. This has already been accomplished to a limited extent by heating the fuel with the exhaust gases.

**148. Semi-Diesel Engines.**—Semi-Diesel engines operate on the Otto cycle with constant volume admission and use a compression pressure of about half that used in the Diesel engine. They obtain their name because they use a heavy oil fuel, the same as used in the Diesel engine.

Because of their low compression pressure the heat of compression is not sufficient for igniting the fuel; hence the semi-Diesel engine must rely on hot bulbs or hot plates for assisting in the ignition. In starting a torch must be used to heat the hot bulb or hot plate.

In the most recent designs the hot bulb and starting torch have been replaced by electrically heated starting plugs. These plugs are used only for a few minutes when starting up, and after warming up the engine depends upon some heat retaining part for igniting the fuel. This gives a safer and more satisfactory engine than those using starting torches.

The advantage of this type of engine is its ability to burn Diesel

TABLE XII.—DIMENSION COMPARISON OF WORLD'S LARGEST MOTORSHIPS

(From *Motorship*, New York)
(The vessels below are in actual service)

| | "*Zoppot*" (Germ. Amer.) | "*Maumee*" (American) | "*Glenogle*" (British) | "*Afrika*" (Danish) | "*Glenapp*" (British) | "*Cubore*" (American) |
|---|---|---|---|---|---|---|
| Displacement (loaded) | 22,000 tons | 15,000 tons | 19,000 tons (abt) | 18,600 tons | 19,000 tons | 17,000 tons (abt) |
| Dead-weight capacity | 17,000 tons | 10,000 tons | 14,000 tons | 13,250 tons | 9,600 tons & 1,700 troops | 11,500 tons |
| Cubic capacity | 693,329 cu. ft. | Not available | Not available | 872,300 | Not available | Not available |
| Length (O.A.) | 545 ft. 0 in. | Not available | 502 ft. 0 in. | 464 ft. 4 in. | 470 ft. 0 in. | 469 ft. 0 in. |
| Length (B.P.) | 525 ft. 0 in. | 455 ft. 0 in. | 485 ft. 0 in. | 445 ft. 0 in. | 450 ft. 5 in. | 450 ft. 0 in. |
| Breadth | 66 ft. 3 in. | 56 ft. 0 in. | 62 ft. 2 in. | 60 ft. 0 in. | 55 ft. 8 in. | 57 ft. 0 in. |
| Depth | 41 ft. 3 in. | Not available | 27 ft. 5 in. | 42 ft. 0 in. | 40 ft. 0 in. | 37 ft. 0 in. |
| Draft (loaded) | 27 ft. 9 in. | 26 ft. 0 in. | 27 ft. 2 in. | 32 ft. 0 in. | Not available | Not available |
| Power | 4,000 i.h.p. | 6,400 i.h.p. | 6,600 i.h.p. | 4,000 i.h.p. | 6,600 i.h.p. | 3,200 i.h.p. |
| Twin or single screw | Twin | Twin | Twin | Twin | Twin | Single |
| Type of Diesel engine | Two cycle | Two cycle | Four cycle | Four cycle | Four cycle | Two cycle |
| Loaded speed | 11½ knots | 14 knots | 13 to 14 knots | 12 knots | 15 knots | 11½ knots |
| Trial speed | 12 1/5 knots | Not available | Not available | 13¼ knots | Not available | Not available |
| Gross tonnage | 9,700 tons | Not available | 9,150 tons | 9,050 tons | 7,263 tons | 7,000 tons |
| Net tonnage | 5,700 tons | Not available | Not available | 5,468 tons | 4,623 tons | Not available |
| Daily fuel consumption | 12 to 13 tons | 20 tons | 20 tons | 13 tons | 20 tons | 17 tons |

engine fuels and the absence of the bad features of the Diesel engine caused by the high compression pressures. The great objection to the engine is the methods required for starting when cold.

Semi-Diesel engines are generally built in small powers up to about 500 h.p. and are very reliable and satisfactory for small craft and auxiliaries where small power is desired.

Among the best known makes of semi-Diesel engines are the Bolinders, Price-Rathbun, Beardmore and Vickers-Peters. The Doxford engine recently brought out in powers up to 3,000 I.H.P. uses a low compression pressure of 300 lbs. and has pistons with thick "heat retaining" heads. The engine is started by steam and uses solid injection with an oil pressure of 9,000 lbs. per square inch. This engine is essentially of the semi-Diesel type.

**149. The Still Engine.**—This is a combination Diesel and steam engine, devised to increase the thermal efficiency over that of the Diesel engine. The heat ordinarily rejected in the jacket water and to the exhaust is used to produce steam and about 8 per cent of this heat is converted into useful work, increasing the B.H.P. of the engine about 30 per cent.

A diagram of the Still engine is shown in Fig. 85. In its present form the engine is double acting with the Diesel cycle working on top of the piston and the steam cycle below the piston. The water jacket is connected in a circuit with the boiler and an exhaust generator as shown in Fig. 85. The cooling water enters and leaves the jacket at a constant temperature corresponding to the pressure of the steam in the boiler. The heat absorbed by the jacket water surrounding the combustion cylinder is used to convert the water into steam at constant temperature. In other words, the heat of combustion that radiates through the walls is transferred into latent heat of steam ($r_1$). The steam thus generated passes to the boiler. The function of the boiler is to produce steam for warming up and starting the engine and to augment the supply generated in the jackets if the jacket supply is not sufficient for the steam end of the cylinder.

The boiler feed water is circulated through the jacket as shown in Fig. 85. The feed water is taken from the feed tank by the feed pump as in all steam plants and is delivered at about 100°

to a feed heater or exhaust generator where it absorbs the heat in the exhaust gases. The temperature of the feed water is thus raised from 100°F. to between 350 and 450°F. and the exhaust gases are reduced from 900 to 150°F. The feed water then enters the jacket where it is converted into steam by the heat of combustion.

The steam from the boiler enters the lower part of the cylinder and acts on the piston in practically the same manner as

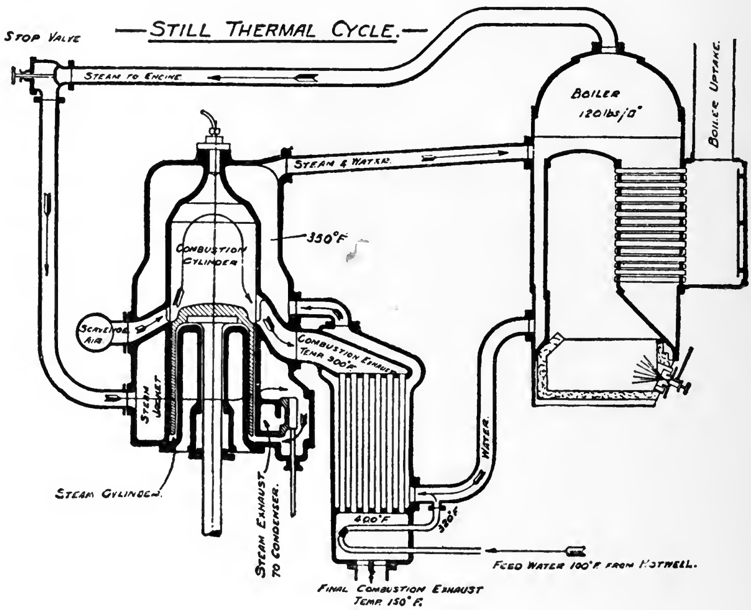



FIG. 85.—Diagram of Still engine.

in a steam engine and is then exhausted to the condenser. Cylinder condensation, which is a large loss with the ordinary steam engine, is practically eliminated in the Still engine because of the heat received from the combustion of the gases in Diesel end.

During compression of the air on the Diesel side of the piston the air charge absorbs heat from the cylinder walls because of the high temperature in the jacket. With the straight Diesel engine the transfer is in the opposite direction, due to the cold circulating water. One result of this is that the required com-

pression pressure is less in the Still engine than in ordinary Diesel engines.

The advantages due to the interaction of the combustion and steam cycles are summarized by F. E. D. Acland in a paper before the Royal Society of Arts as follows:

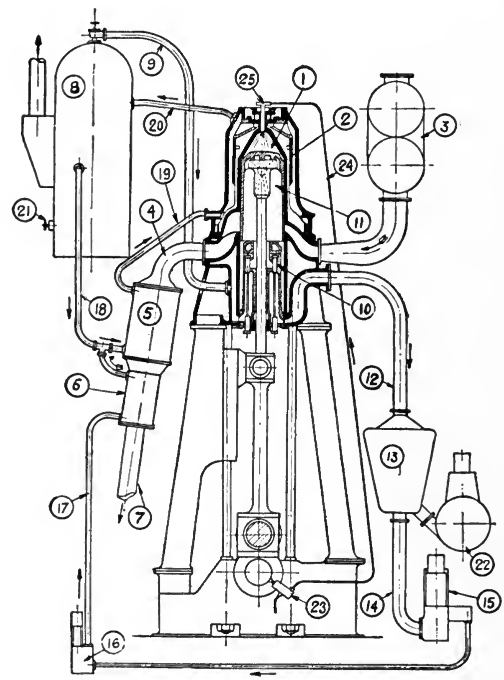
1. The mean temperature of the cylinder walls is higher than in ordinary engines, the cooler parts being maintained at a higher, the hotter parts at a lower temperature.
2. The piston is cooler, owing to the expansion of the steam behind it.
3. The heat efficiency of the combustion cycle is augmented owing to the walls being at a higher and constant temperature, and is in proportion to the rise in temperature of the jacket water.
4. Frictional losses are reduced by the higher temperature, and by the steam overcoming the inertia of the reciprocating masses at the lower dead center.
5. The mechanical efficiency of the whole engine is higher than that obtainable in a normal engine of similar type.
6. The steam, expanding as it does in a cylinder hotter than itself, gives an indicator diagram larger than that theoretically obtainable under ideal conditions in an ordinary steam engine.
7. Twenty-nine per cent of additional brake horsepower is added to the shaft of the engine, without increase in the fuel consumption. (Steam not condensed.)
8. Forty per cent is added with condenser used. (Air pump separately driven.)
9. The I.H.P. due to steam appears as B.H.P. added to the shaft, all the mechanical losses having already been accounted for in measuring the combustion B.H.P.

Besides the merits listed above the two-cycle Still engine has the following advantages:

1. Fuel consumption 10 to 20 per cent lower than the Diesel engine.
2. Absence of cold circulating water causing large temperature difference and trouble with cylinder and head castings.
3. Absence of cylinder condensation in steam end.
4. Lower compression pressure.
5. Absence of air starting, circulating, and piston cooling system.
6. Absence of exhaust valves and gear.
7. Increased horsepower for a given bore and stroke.
8. Possibility of overload by forcing steam boiler.
9. Maneuvering at low r.p.m. is possible.
10. High temperature range 2,000° to 150°F. (Carnot efficiency).

Trials with a single cylinder two-cycle Still engine were carried out during 1921 by Scott's Shipbuilding & Engineering

Co., Greenock, Scotland. Owing to the fact that the experimental engine had only one cylinder, it was necessary to provide a small auxiliary H.P. steam cylinder, the lower end of the main Still cylinder serving as a L.P. steam cylinder. In an actual installation where several cylinders are used the lower part of




1. Combustion Cylinder.
2. Reinforcing Steel Hoop.
3. Scavenging Blower.
4. Combustion Exhaust Pipe Jacketed by Boiler Water.
5. Exhaust Generator.
6. Feed Water Heater.
7. Fina Combustion Exhaust to Atmosphere.
8. Boiler.
9. Main Steam Pipe.
10. Steam Inlet and Exhaust Valves.
11. Steam Cylinder.
12. Steam Exhaust to Condenser.
13. Condenser.
14. Suction Pipe to Air Pump.
15. Air Pump.
16. Feed Pump.
17. Delivery Pipe to Feed Heater.
18. Circulating Water, Boiler to Exhaust Generator.
19. Circulating Water, Exhaust Generator to Cylinder Jacket.
20. Circulating Water and Steam; Jacket to Boiler.
21. Auxiliary Oil Burner.
22. Condenser Circulating Pump.
23. Oil Fuel Injection Pump.
24. Oil Fuel Delivery to Injection Valve.
25. Injection Valve.

Fig. 86.—Section through Still engine.

one can be used as a H.P. cylinder and the lower part of another as a L.P. cylinder, thus obviating the use of an auxiliary cylinder. All the auxiliaries except the scavenging air pump were driven off the main engine in these trials. A diagrammatic view of this

engine and all the auxiliaries is shown in Fig. 86. The results of the trials of this engine are given in the following table:

Trials of 22 by 36-in. Still Oil Engines

| Main Still cylinder | Auxiliary H.P. cylinder |
|---|---|
| Stroke 36 in. | Stroke 14 in. |
| Bore 22 in. | Bore 22 in. |
| Piston rod $6\frac{1}{4}$ in. | |

| Item | | Overload | Full load | Half load |
|---|---|---|---|---|
| 1 | Average combustion m.e.p. lbs./sq. in. | 88.9 | 81.2 | 54.2 |
| 2 | Average steam mep referred H.P. cyl. | 4.43 | 3.80 | 1.26 |
| 3 | Average steam m.e.p. referred L.P. cyl. | 7.36 | 6.23 | 3.60 |
| 4 | Total m.e.p. | 100.69 | 91.25 | 59.06 |
| 5 | R.p.m. | 128.1 | 124.3 | 103 |
| 6 | Steam boiler pressure, lbs./sq. in. gage | 112 | 100 | 108 |
| 7 | H.P. receiver, lbs./sq. in. gage | 75 | 57 | 23.5 |
| 8 | L.P. receiver, lbs./sq. in. gage | 11 | 5.5 | 0.4 |
| 9 | Vacuum, inches hg | 28 | 27.5 | 26.6 |
| 10 | Water evaporated per hour, lbs | 950 | 807 | 388 |
| 11 | Scavenging pressure, inches water | 49 | 46 | 40 |
| 12 | B.H.P. for scavenging | 15.4 | 14.1 | 12.0 |
| 13 | Combustion I.H.P. | 394 | 349.5 | 192.5 |
| 14 | Total I.H.P | 446 | 392 | 210 |
| 15 | Engine B.H.P. | 384 | 343 | 174.5 |
| 16 | Net b.h.p. (Item 15–Item 12) | 368.6 | 329 | 162.5 |
| 17 | Oil per hour, lbs | 146.6 | 123.4 | 64.0 |
| 18 | Oil per net B.H.P. per hour | 0.398 | 0.375 | 0.394 |
| 19 | Efficiency on net B.H.P., per cent | 35.5 | 37.7 | 35.8 |

It will be observed that the fuel consumption at full load is .375 lb./B.H.P., which is about 10 per cent lower than the best four-cycle Diesel practice and nearly 20 per cent better than the general run of two-cycle Diesel engines.

Claims are put forward that the Still engine weighs less and occupies less space than the Diesel engine. The gain in economy of between 10 and 15 per cent for this single cylinder engine is highly encouraging and no doubt this can be improved upon when several cylinders are used. At first thought the engine appears complicated but it must be borne in mind that the air starting, circulating and piston cooling systems are

eliminated and the small boiler employed with the Still engine would generally be required on a Diesel ship for heating purposes.

It would seem that the claims of the inventor are borne out by the recent trials and the engine has excellent possibilities for marine propulsion.

## CHAPTER XIV

## COMPARISON OF TYPES OF PROPELLING MACHINERY

**150. General.**—Considerable difference of opinion exists today on the proper type of machinery to use for ship propulsion. The installation to give the most economical performance in service does not necessarily mean the one showing the lowest fuel consumption, for hand in hand with a study of fuel and steam consumption must go a study of capital charges, depreciation, cost of maintenance, weight, space occupied, r.p.m. and propeller efficiency, possibilities of breakdown and the general feasibility and reliability of the installation as a whole. As an example, a case is on record where a quadruple expansion (land installation) showing excellent fuel consumption was replaced by another engine because of the high overall maintenance cost. Here is a case where the low fuel consumption was outweighed in actual practice by other considerations.

Another point that should not be overlooked is the interrelation of the efficiency of the power plant with the operating efficiency of the ship as a whole. Obviously, if the repairs or overhauling of a particular unit of a ship's power plant, or the ability to obtain fuel, prolonged the ship's stay in port over that necessary for discharging and loading the cargo, it would be poor judgment to install such a unit or adopt a particular fuel, no matter what gain in fuel economy resulted. For a slow or medium speed cargo ship the fuel cost is not a large proportion of the total operating expenses, and hence a reduction in fuel consumption should not be accomplished at the expense of some more important factor in the operating charges.

The power plant and fuel have another relation to the ship as a whole, in that they both affect the cubic feet of cargo space, the tonnage measurement, and the net deadweight capacity. We should not lose sight of the fact that a reduction in steam consumption is accompanied by a reduction in the size (and cost) of the boilers, piping, auxiliaries and bunkers which

in turn increase the carrying capacity of the ship. So complex is the problem that no exact solution is possible—the best that can be done is to give proper weight to all the factors involved.

The tendency in the past has been too conservative in regard to innovations and improvements in a ship's power plant. There has been some justification for this, for if troubles arise with new types of machinery at sea the engineer cannot call on the manufacturer for help and assistance as in a land installation, but must make his own repairs and solve his own problems. Further, when a ship is at sea the superintending engineer does not have access to the plant as in a land installation and the chief engineer is thrown absolutely on his own resources. A breakdown of a ship's machinery at sea is always a more serious affair than in a power plant on shore; at times it might seriously endanger the ship. During the war marked advances were made in marine engineering and much of the conservatism has been swept aside and many things almost impossible a few years ago are today larger adopted. Still, there is a great deal of conservatism left among shipowners, as the failure to adopt Diesel engines and the continued use of the Scotch boiler engines bear witness.

**151. Types.**—In the selection of machinery for a merchant ship, we have the following types to choose from:

1. Triple expansion reciprocating engines.
2. Quadruple expansion reciprocating engines.
3. Direct connected turbines.
4. Combination machinery—triple expansion engines and L.P. turbine.
5. Geared turbines.
   Single reduction 1:5 to 1:10.
   Double reduction 1:25 to 1:35.
6. Electric drive.
7. Diesel engines.
   (*a*) Two- or four-cycle.
   (*b*) Solid injection; air injection.
8. Diesel electric drive.
9. Still engine—combination of steam and Diesel cycles.

All of these types have virtues and defects and a selection cannot be made on a few points alone.

The important factors to be considered in a selection are:

1. R.p.m. and propeller efficiency.
2. Fuel consumption.

3. Weight, and space occupied.
4. Capital charges, including maintenance, repairs and depreciation.
5. General reliability.
6. Time required in port for overhauling and repairs.

For a low speed cargo ship of moderate power, high propeller efficiency requires low r.p.m. With the exception of the direct connected turbine, all the types mentioned can be designed for revolutions from 80 to 130.

With direct connected turbines the high r.p.m. necessary for good turbine efficiency, reasonable weight, and size are detrimental to propeller efficiency; hence this type of machinery has practically disappeared from ship propulsion and will not be considered here. The last noteworthy installation was on the S.S. "Lusitania" and S.S. "Mauretania," which were ships of 25 knots speed with the propeller running at 180 r.p.m.

**152. The Reciprocating Engine.**—With the exception of the four-cycle Diesel engine the reciprocating engine as represented by the three and four cylinder triple expansion and the quadruple expansion types is the heaviest and most bulky prime mover used for ship propulsion. The marine reciprocating engine always shows a higher steam consumption than the turbine. This is due partly to the lower steam pressure, lower superheat and lower vacuum used with the reciprocating engine and partly to the lower efficiency ratio which is inherent with the reciprocating engine. It was pointed out in Art. **90** that it was inadvisable to carry more than 27 or 27 1/2 in. of vacuum with a reciprocating engine while with a turbine 29 in. is possible. Reciprocating engines could, however, carry as high a steam pressure and superheat as is used with turbines, but generally when the former type of machinery is used the same conservatism that leads to the adoption of the reciprocating engine results in the use of lower steam pressures and generally no superheat. We thus find the turbine working between more extreme limits and hence having a larger Rankine efficiency.

The efficiency ratio of the reciprocating engine is lower than a turbine even when both prime movers are working between the same limits. This is due to the large losses by condensation and leakage in the reciprocating engine. The engines of the U.S.S. "Delaware," which were exceptionally well designed engines, had an efficiency ratio of only .55; and the S.S. "Rampo"

with a quadruple expansion merchant engine showed an efficiency ratio of only .597. Geared turbines, on the other hand, will invariably show efficiency ratios of .60 and .65 (see Art. **184**). Further, the turbine will continue to show good economy after long service while the efficiency of the reciprocating engine will fall off rapidly due to the wear of the numerous moving parts such as pistons, valves, cylinders, cylinder liners, etc.

Some land installations with low r.p.m., large ratio of expansion, and fitted with jackets, reheaters, etc., have shown excellent economy as indicated in Table IX. This table shows efficiency ratios as high as .70 and .74 for stationary reciprocating engines. Such units would not be at all feasible on shipboard on account of weight and space requirement. If steam and fuel consumption, weight and space were our only criterion in selecting machinery the reciprocating engine would receive little consideration today. The triple expansion engine, however, is an installation of proven reliability, repairs can generally be made in any part of the world, and marine engineers are familiar with the unit. The geared turbine and electric drive, on the other hand, are new and in some respects untried installations. Repair bills and the time lost in overhauling these units when operated by inexperienced crews may more than offset the advantages of lower steam consumption and reduced machinery space.

The four cylinder triple expansion engine with the improvements in design such as were used in the last reciprocating engined ships of the U. S. Navy has much to recommend it for certain low powered cargo ships, such as tramps, on account of the character of the engine room force often available and possibilities of repairs in out-of-the-way ports. There is no doubt, however, that too many reciprocating steam engines are being installed today, especially with the advantages offered by the Diesel engine. The quadruple expansion engine shows better economy than the triple expansion engine, but its first cost is greater and it requires a longer fore and aft space. Thus, while there is a reduction in annual coal bills and a reduction in bunker and boiler space, there is a loss of cargo space on account of the longer engine room required and an increase in capital charges due to greater first cost of the engine. Unless the ship is of sufficient size and the voyage long enough to show a fair increase

in annual profits due to reduction in fuel costs and bunker and boiler space as offset against increased capital charges and loss of cargo space (due to length of engine), the quadruple expansion engine is not to be recommended. As a rule the four cylinder triple expansion engine is to be preferred in place of a quadruple expansion engine.

**153. Combination Machinery.**—Combination machinery at one time offered great possibilities in reduced steam consumption and would no doubt have been widely adopted except for the introduction of the geared turbine. This type of machinery consists of a three screw arrangement with reciprocating engines on the wing shaft, exhausting to a low pressure turbine on the center shaft. The steam consumption of this arrangement is nearly as good as that of the geared turbine. This is due to the high efficiency of the reciprocating engine at high pressures and the turbine and low pressures. The noteworthy installations are the S.S. "Otaki," S.S. "Olympic" and S.S. "Rochambeau." The machinery requires a larger space than reciprocating engines because of the necessity of placing the turbine on the center shaft aft of the wing engines. It was found in the "Otaki" installation, however, that the loss in cargo space was offset by the reduction in bunker and boiler space. There resulted a considerable net gain due to reduced coal consumption. The "Minnekahda" of the Red Star line, built as late as 1917, is fitted with this type of machinery.

**154. Geared Turbine and Electric Drive.**—From the foregoing it appears that the choice of a steam installation narrows down to the geared turbine and the electric drive. These two types of machinery do not differ greatly from one another in steam consumption, weight, and space occupied. Certainly, the small difference in these factors is not enough to have an influence in the selection of one or the other for a cargo ship. Both can reduce the propeller revolutions down to a reasonable value. The little difference existing is in favor of the geared turbine which can reduce the r.p.m. down to a value as low as is desired while the lower limit for a motor of less than 3,000 S.H.P. is about 100 r.p.m.[1] The efficiency is also slightly in favor of the geared turbine which has an efficiency of 95 per cent for the double reduction gears against 91.5 per cent for the combined efficiency of the generator and motor. However, there is a

small loss due to windage in the reversing turbine of the geared installation which reduces the output of this installation. The net result is that the geared installation requires about 2 per cent less shaft horsepower than electric drive.[1] To this should be added a slight possible gain in propeller efficiency if lower r.p.m. are adopted with the geared turbine.

Ease of control and reversing, and economy at reduced speeds are greatly in favor of the electric drive. Also, due to the fact that the electric drive requires no reversing element, a higher initial steam temperature can be carried which allows greater possibilities in economy by using higher steam pressures and superheat. Because of the constant speed inherit with the synchronous motor, racing of the propeller in heavy weather is impossible with the electric drive when synchronous motors are used. Ships equipped with this type of machinery have shown almost constant r.p.m. in a heavy sea. For this reason the ship fitted with electric drive would show better economy and outdistance the geared turbine ship in heavy weather.

The electric drive has been proven beyond all dispute to be the ideal machinery for naval vessels where ease of handling during maneuvering is essential and where good economy at cruising speeds is required. These merits, however, do not appear to be pertinent for merchant ships where continuous duty at full power and a minimum amount of maneuvering are involved.

The claim has been put forward for the electric drive that it eliminates the shaft alley, but this is only accomplished by a reduction in cargo space caused by the motor compartment aft. It is also very poor practice to place a piece of machinery in a compartment that has no communication with the engine room except by going on deck.

The electric drive has one advantage over the geared turbine that may eventually prove of more importance than it does today—namely, the life of the installation. This turbine will last indefinitely and will require little attention except for the bearings, but the gears will probably have to be replaced from time to time due to wearing, pitting, etc. The electric machinery, however, will probably outlast several sets of gears. This may prove an important factor in operating costs.

[1] *Mar. Eng.*, Dec., 1919, p. 817.

During the haste and rush of the war a number of geared installations fell down due to poor workmanship; others failed because of incorrect design caused by a lack of experience. Perhaps the largest amount of trouble, and that which has caused such a lot of prejudice and hesitation about adopting geared turbines, has been due to the inefficient handling on shipboard by inexperienced crews.

With the careful workmanship which is available today, extended experience in design, and proper attention to the oiling system after installation, there should be little trouble experienced with this installation. Most of the merchant ships fitted with double reduction gears today are operating perfectly. A paper before the Institution of Naval Architects in 1920 by H. B. Tostevin points out that during the war there were 556 sets of geared turbines in the British Navy, some having been in operation 6 years. Of these, three were removed for realignment; two were removed and "dressed up" because of noise and slight wearing; two others developed fractured teeth but were not removed. Surely this is a creditable record.

The ships of the British Navy just referred to are practically all fitted with single reduction gearing with reductions between 1 : 5 and 1 : 10, the propeller running at 200 to 400 r.p.m. While revolutions as high as this give good propeller efficiency with high speed naval vessels, they are not admissible for slow speed merchant ships. Hence double reduction gearing has been adopted in the merchant service.

A fair comparison, however, shows that the geared turbine and electric drive are on a par in most respects and are by far the best choice for steam machinery. If the cost of the electric drive can be made the same as the geared installation probably both will be used in the future.

The electric drive in connection with the Diesel engine offers special advantages for small installations and will be taken up later in this chapter.

As already pointed out, the lower the propeller r.p.m. the better the efficiency of the propeller. For cargo ships of 10 and 11 knots speed with about 3,000 S.H.P. r.p.m., of about 90 have been adopted with geared turbines. Revolutions lower than this are even more desirable if suitable gearing can be secured and the propeller diameter does not become too exces-

sive. We thus see that the introduction of the geared turbine offers low propeller r.p.m. coupled with light weight as well as low steam consumption.

**155. Space Occupied by Machinery.**—The adoption of improved types of machinery and consequent reduction in the machinery space may lead to a reduction in revenue, and possibly a loss, at a place entirely unlooked for—namely, in the port and canal dues which are charged against net tonnage. The existing rules (1921) for net tonnage measurement allow a reduction from the gross tonnage of 32 per cent for the machinery space, provided the actual measurement of this space is above 13 and under 20 per cent of the gross tonnage. If this space does not lie between these limits the reduction is 1.75 times the actual space. With improved machinery and reduced space the percentage might come out 12 per cent. In this case the rule allows $12 \times 1.75$ or 21 per cent reduction instead of 32 per cent.

A. T. Wall, who first directed attention to this fact in a paper before the Institution of Naval Architects in 1919, cites the following example: Ship $580 \times 70 \times 45$ ft. with a gross tonnage of 20,000 and fitted with reciprocating engines and Scotch boilers. The space occupied by engines and boilers works out 14 per cent, allowing a reduction from the gross tonnage of 32 per cent, or 6,400 tons. With an allowance of 1,350 tons for crew space, the total reduction becomes 7,750 tons. This gives a net tonnage of 12,250 tons. If the same ship is fitted with turbine machinery and water-tube boilers the actual space is 10 per cent, which allows a reduction of $10 \times 1.75$ or 17.5 per cent instead of 32 per cent as before. This results in a net tonnage of 15,150—an increase in net tonnage of 24 per cent with an actual gain of only 5 per cent in cargo space.

Thus the owner is required to pay tonnage dues on 2,900 additional tonnage by the adoption of more efficient machinery. While the present tonnage measurements are in force it appears that the reduction in space brought about by turbine engines and water-tube boilers might result in a loss instead of a gain.

This again illustrates the point previously mentioned—that there are many factors which involve the hull as well as the machinery installation and that a decision as to the most suit-

able power plant cannot be given until every influencing factor is given its proper weight.

The geared turbine occupies considerably less space than the reciprocating engine and about the same space as the electric drive. A careful comparison recently carried out by the author between (1) a 3,200 I.H.P. single screw installation consisting of a reciprocating engine, Scotch boilers and coal fuel; and (2) a 2,950 S.H.P. installation consisting of geared turbine water-tube boilers and fuel oil, showed that the fore and aft space occupied by machinery and fuel (including a cross bunker) in the former installation was practically twice that occupied by the latter. The S.H.P. of the geared turbine was fixed at 92 per cent of the I.H.P. of the reciprocating engine, and the fuel radius was taken as 4,000 miles.[1]

As an illustration of the small space occupied by geared turbine machinery, attention is directed to the S.S. "Andrea Luckenbach,"[2] which consists of a twin screw geared turbine and water-tube installation of 6,000 S.H.P. The fore and aft space occupied by the machinery is only 49 ft. No reciprocating or Diesel engine installation of similar power could be accommodated in this small space.

**156. The Diesel Engine.**—The superior fuel economy of the Diesel engine over all other types of prime movers is very marked. The fuel consumption of this engine averages about 0.42 lb./S.H.P., while the very best performance with geared turbines and water-tube boilers is about 0.90 lb./S.H.P. per hour. With the adoption of high steam pressures, superheat, high vacuum, and the most efficient, oil burning water-tube boilers and geared turbines, together with the most careful attention to the economical operation of the whole power plant, the fuel consumption for a steamship might be reduced to 0.80 lb./S.H.P. A consumption as low as this, however, would be practically impossible to maintain for any length of time in service conditions at sea. The best that can be expected of the steam installation in service is probably about 1.00 lb. per S.H.P. per hour. The Diesel engine, however, can easily maintain a consumption of 0.42 lb./S.H.P. day in and day out at sea. Hence we can conclude that the

[1] See art. 162.

[2] An arrangement drawing of this ship is shown in Plate IX of *The Shipbuilding Cyclopaedia.*

best steam installation will have a fuel consumption of between 2 and 2.5 times that of the Diesel engine. This is comparing the very best type of steam machinery with the Diesel engine. If we extend the comparison to coal burning ships fitted with reciprocating engines the fuel used will run from 3 to 3.5 times that of the Diesel engine.

The Diesel engine has now reached the stage in its development where shaft horsepowers of 2,500 to 3,000 are in successful operation. With twin screws, an installation of 6,000 S.H.P. per ship is possible today. The revolutions of Diesel engines now in operation in cargo ships range from 85 to 130.

Diesel machinery has the disadvantage when compared with steam machinery, in that it is considerably heavier and has a higher first cost. It has the advantage in that it has a fuel consumption of 1/4 to 1/2 of that of steam installations and the stand-by losses are practically eliminated. A geared turbine installation with water-tube boilers will occupy less space than a four-cycle Diesel engine, and about the same space as a two-cycle engine.

Unfair comparisons of Diesel and steam machinery have at times been published showing that the Diesel machinery is no heavier and cost no more per horsepower than steam. These comparisons, on closer examination, generally reveal the fact that a Diesel engine with 120 or more r.p.m. is compared to a steam installation of 70 to 90 revolutions. All comparisons of cost and weight should, of course, be made at the same r.p.m.

If an actual comparison of ships on a given trade route is made, and the operating expenses and profits calculated, the advantages of the motorship generally offset the small disadvantages of higher first cost and greater weight per horsepower provided the motorship does not have to carry fuel for the round trip and the price of Diesel engine fuel is the same as "fuel oil."

The objections raised to Diesel machinery by vessel owners appear to be mostly imaginary; it is a new type of machinery still in a state of development and subject to breakdowns; repairs may be difficult to obtain for this new type of machinery, causing interrupted service and long delays in port with consequent loss in profits; Diesel machinery is easily manhandled in the hands of an inexperienced crew; there is a scarcity of operating

engineers and if the Diesel engine is widely adopted ships may be held up for lack of a trained engine room force; breakdowns and various troubles are frequently reported, and oil cannot be secured in all ports.

Along with these objections we must bear in mind that men trained in steam engineering hesitate to give up a well known and reliable type of machinery for a new and (to them) precarious type. New types of machinery have always been slowly adopted on shipboard as the operating engineer must be his own repair man and the engineer cannot call on the manufacturer for advice and assistance as could be done at a power plant on shore. This last is a strong argument against the rapid adoption of the Diesel engine for ship propulsion and can only be overcome by providing well trained and competent operating engineers. Careful attention on the part of the engine builders to see that the crew is properly instructed regarding the operation of their engine and informed of all troubles likely to occur, will go a long way to solving this problem.

The other objections can be thrown aside because the successful motorship is a demonstrated fact. Those who are competent to answer assure us that well trained Diesel engineers are available at all times.

The Diesel engine on account of its small power and large weight per horsepower cannot be adopted at present for high powered ships or fast passenger vessels. Also for the same reason it is not suitable for naval vessels. For the cargo ship and small passenger liners the Diesel engine is an ideal installation. Its adoption will generally reduce the operating expenses on account of the reduced fuel costs and increase the annual profits because of the greater carrying capacity brought about by the large reduction in fuel weight and space required for fuel bunkers. The weight and space occupied by fuel for a Diesel engine is about 25 per cent of that required for a coal burning and 40 per cent of that for an oil burning steamship (see Chap. VII).

With a short voyage, long port detention or slow speed, the gain of the motorship is not so pronounced over that of the steamship on account of the small percentage that the fuel costs bear to the total operating expenses. It is the combination of reduced fuel costs and increased deadweight coupled with quick

turn arounds and large number of trips per year that produce the large gain in profits.

In comparing the fuel consumptions of Diesel engines and oil burning steamships it must be remembered that the most makes of Diesel engines require a lighter and more expensive grade of fuel than steamship "fuel oil." This lighter Diesel oil is more difficult to procure and is not available in all ports. The price of Diesel fuel at times is 50 per cent dearer than "fuel oil." This fact is important and will cut down to a considerable extent the gain of the Diesel engine over the steam installation. However, this is probably only a temporary condition that will pass with a more uniform demand for Diesel fuel and the improvement in Diesel engines so that the heavy and inferior oils can be used. As already pointed out (Art. **147**), the two-cycle port scavenging type lends itself more readily to the use of the heavier oils and for this reason will probably be used to a large extent in the future.

**157. Comparison of Two- and Four-Cycle Diesel Engines.**—The two-cycle Diesel engine is a far more attractive proposition on paper than the four-cycle, but on the other hand is not so well known nor so universally used as the four-cycle. It appears that the real question is not so much a comparison or selection of cycle as of make of engine. The two-cycle engine is more difficult to design and is not so well understood and there have been more frequent breakdowns and troubles with this engine than with the four-cycle. Hence those arguing for the four-cycle engine have laid all these misfortunes at the door of the two-cycle in general instead of the manufacturer in particular. There are some builders of two-cycle engines, Sulzer Bros. for example, who have had very good success with two-cycle engines and apparently have made good most of the claims put forward for this type.

Most of the attractive features of the two-cycle engine are not important enough to warrant the installation of this type in a cargo ship if there is more probability of trouble than with a four-cycle engine. The loss of one trip per year or increased port detention due to frequent repairs will quickly offset all the small gains brought about by more economical machinery.

Many of the features of the two-cycle engine are advantageous for passenger ships and ships where weight of machinery is an

important item. No doubt the two-cycle engine of the port scavenging type is destined to have a large field in the future, especially after further development and experience.

The advantages claimed for the two-cycle engine are:

1. Weight 60 to 65 per cent of four-cycle at same r.p.m. (hence reduced cost).
2. Space 55 to 80 per cent of four-cycle at same r.p.m.
3. With port scavenging, less complicated head casting and simpler valve gear.
4. More uniform turning moment; reduced vibrations; shorter crankshaft.
5. Absence of exhaust valves.
6. Ability to operate on cheaper and inferior fuels.

The disadvantages charged against the two-cycle engine are:

1. Higher fuel consumption (5 to 10 per cent).
2. Higher lubricating oil consumption.
3. Lower m.e.p.
4. Special means of cooling required on account of great heat effects.
5. On account of heat, piston cooling must be adopted at high r.p.m.

As pointed out in Chap. XIII there are some makes of two-cycle engines, notably the Sulzer Bros., to which most of these objections do not hold.

**158. Diesel-Electric Drive.**—The Diesel electric drive has been frequently proposed for merchant ships and to some extent actually fitted in small powers. The merits of this type of machinery are that one or more high speed Diesel engines driving electric generators can be installed and the ship propelled by an electric motor of low r.p.m., and hence high propeller efficiency obtained without reducing the revolutions of the engine.

Most of the claims brought forward for the electric drive with steam machinery apply equally well to this type of installation and, as already pointed out, offer no particular attractions for cargo ships.

For the electric drive the first cost may be slightly lower than for direct Diesel drive; the space occupied about the same; the saving in weight will be small; and the installation will be much more complicated. Certainly little gain in economy or propeller efficiency can be claimed and the direct connected Diesel engine of low r.p.m. will be more reliable in service than the high speed generating sets.

The Diesel-electric drive, however, does offer many advantages

for naval vessels, yachts and fishing vessels where easy control and high economy at reduced speed are important factors.

**159. Geared Diesel Engines.**—High speed Diesel engines with reduction gearing between the engine and propeller have been used to some extent. An example of this is the S.S. "Havelland" which has a twin screw installation consisting of two 10-cylinder 1,650 S.H.P. Diesels running at 230 r.p.m. (800 ft. per min. piston speed) and geared down in a ratio of 2.7 : 1 so that the propeller runs at 85 r.p.m. The engines used in this case were German submarine engines built originally to run at 390 r.p.m.; hence this installation was adopted to make use of engines already on hand. It does not have any advantages over the Diesel-electric drive and many of the valuable features of the latter drive are not obtained by the geared drive.

**160. Semi-Diesel Engines.**—This engine is available up to powers of 500 S.H.P. and because of its lighter weight, cheaper cost and its all-round simplicity, it is an ideal propelling unit for small auxiliary sailing vessels and other craft where the power required does not exceed 300 S.H.P. The semi-Diesel engine is used to a considerable extent for an emergency auxiliary air compressor and generator drive on motorships.

**161. Gasoline Engines.**—This type of internal combustion engine is used only for small launches and motorboats where light weight and small horsepowers are essential. Because of the high cost, and danger in storage and handling of gasoline, this engine is not used in commercial vessels.

**162.—Comparison of Seven Types of Machinery Installation for a Voyage of 4,000 Miles.**—The following comparison has been made for ships fitted with:

1. Reciprocating engine, Scotch boilers and coal.
2. Reciprocating engine, water-tube boilers and oil.
3. Geared turbines, water-tube boilers and oil.
4. Four-cycle Diesel engines, (twin screws).
5. Two-cycle Diesel engines, (twin screws).
6. Diesel-electric drive (single screw).
7. Turbo-electric drive (synchronous motor).

A ship with the following characteristics has been used for each installation:

L.B.P. = 418′–0″.
Beam = 60′–0″.

Depth = 35′–6″.
Load draft = 26′–0″.
Sea speed = 11½ knots.
Load displacement = 14,600 tons.
Block coef. = .78.
Weight of hull and equipment = 4200 tons.
The E.H.P. was calculated from Robertson's data for a longitudinal coefficient of forebody = .83, and aftbody = .77.

Proper allowance has been made for struts on the twin screw ships, for hull efficiencies, changes in wake, and propeller efficiencies, as shown in the following table. The notes below the table should be consulted in this connection.

The machinery spaces in all the ship were greater than 13% of the gross tonnage; hence the gross and net tonnage for all the ships are the same except the coal burning ship which is slightly less than the others as no fuel is carried in the double bottoms.

The table shows some very interesting results and is worthy of careful study. Attention is especially invited to the results shown in lines 23, 24, 30 and 31.

As laid out the turbo-electric drive proves itself superior in space occupied, although the geared turbine and two-cycle Diesel installations occupy practically the same space. The two cycle Diesel installation allows the largest net cargo deadweight, although the difference between the two and four cycle engines in this respect is not large. The four cycle Diesel installation has the lowest fuel consumption.

The geared turbine and water-tube boiler installation is the cheapest, gives nearly the same net cargo deadweight as the Diesel installation, and allows a volume of cargo space practically as large as either the two-cycle Diesel or the turbo-electric drive.

The above comparison has been carried through in an impartial and unbiased manner, the machinery arrangements[1] were carefully laid out from drawings furnished by the engine builders, and the weights summarized from authentic data. No doubt in the hands of other designers the spaces occupied and the machinery weights would vary somewhat; efficiencies and fuel consumptions, however, cannot be varied very much. While variations here and there might be introduced, the results given above are comparable

[1] The machinery arrangements were made by the senior students in the course in Ship Design at Lehigh University under the direct supervision of the author.

and shed considerable light on certain aspects of the types of propelling machinery in use today.

Of course a full and complete comparison must be made of operating costs on a given trade route to determine which is the all around most economical type. If authentic figures are available for the costs of machinery, fuel, lubricating oil, depreciation, repairs, insurance, etc., operating costs can be quickly worked up from the data given in the above table.

**163. Conclusions.**—In reviewing the advantages and disadvantages of the various types of engines used for ship propulsion it is seen that the engineer and shipowner are confronted with a large number of types all of which are in successful commercial operation. No definite decision can be given as to the best type, for so many factors are involved that vary with the ship, the trade route, the cargo to be carried, the type of crew available, etc. Each case must be carefully studied from the viewpoints of efficiency, operating charges and all-round feasibility, and what is the test type for one ship will not be the best for another.

It should be borne in mind that much of the published data and information available has been presented by men advocating a particular type of propelling machinery and the data are often biased and unfairly presented.

TABLE OF COMPARISON OF MACHINERY INSTALLATIONS.

| | Reciprocating Engines, Scotch Boilers, Coal 1 | Reciprocating Engine, W.T. Boilers, Oil 2 | Geared Turbines W.T. Boilers, Oil 3 | 4 Cycle Diesel 4 | 2 Cycle Diesel 5 | Diesel-electric 6 | Turbo-electric Synchronous Motor 7 |
|---|---|---|---|---|---|---|---|
| 1. Model E.H.P. +10% | 1,720 | 1,720 | 1,720 | 1,760 | 1,760 | 1,720 | 1,720 |
| 2. No. of screws[1] | 1 | 1 | 1 | 2 | 2 | 1 | 1 |
| 3. Propeller r.p.m. | 75 | 75 | 90 | 115 | 115 | 90 | 90 |
| 4. Propeller efficiency[2] | .585 | .585 | .550 | .605 | .605 | .550 | .550 |
| 5. Propulsive coef.[3] | .615 | .615 | .577 | .605 | .605 | .577 | .577 |
| 6. I.H.P. | 3,040 | 3,040 | ...... | ...... | ...... | ...... | ...... |
| 7. S. H. P. delivered to propeller | 2,800 | 2,800 | 2,980 | 2,910 | 2,910 | 2,980 | 2,980 |
| 8. S.H.P. of prime mover | 2,800 | 2,800 | 2,980 | 2,910 | 2,910 | 3,600 | 3,240 |
| 9. Boiler pressure (gage) | 200 | 250 | 250 | ...... | ...... | ...... | 250 |
| 10. Superheat, — °F. | 60° | 60° | 100° | ...... | ...... | ...... | 150° |
| 11. Vacuum, — inches of mercury | 27 | 27 | 28½ | ...... | ...... | ...... | 28½ |
| 12. $H_1 - q_2$ (Feed = 218°) | 1,041 | 1,057 | 1,079 | ...... | ...... | ...... | 1,103 |
| 13. $H_1 - H_2$ | 332 | 347 | 388 | ...... | ...... | ...... | 403 |
| 14. Efficiency ratio[4] | .59 | .59 | .63 | ...... | ...... | ...... | .67 |
| 15. Steam used by main engine (lbs. per S.H.P. per hour.)[5] | 14.15 | 13.50 | 10.40 | ...... | ...... | ...... | 9.40 |
| 16. Auxiliary steam (% of main engine's) | 15% | 15% | 25% | ...... | ...... | ...... | 25% |
| 17. Total steam consumption (lbs./S.H.P. /hr.) | 16.25 | 15.50 | 13.00 | ...... | ...... | ...... | 11.75 |
| 18. Total steam per hour | 45,500 | 43,400 | 38,800 | ...... | ...... | ...... | 38,100 |

TABLE OF COMPARISON OF MACHINERY INSTALLATIONS—*Continued.*

| | Reciprocating Engines, Scotch Boilers, Coal 1 | Reciprocating Engine, W.T. Boilers, Oil 2 | Geared Turbines W.T. Boilers, Oil 3 | 4 Cycle Diesel 4 | 2 Cycle Diesel 5 | Diesel-electric 6 | Turbo-electric Synchronous Motor 7 |
|---|---|---|---|---|---|---|---|
| 19. Rate of combustion (lbs./sq.ft.H.S./hr.) | .55[6] | .55 | .55 | ...... | ...... | ...... | .55 |
| 20. Boiler efficiency in service[7] | .66 | .72 | .72 | ...... | ...... | ...... | .72 |
| 21. Calculated heating surface+10% | 9,960 | 6,720 | 6,150 | ...... | ...... | ...... | 6,190 |
| 22. Fuel per S.H.P. of prime mover per hour[5] | 1.78 | 1.20 | 1.03 | .42 | .44 | .46 | 1.035 |
| 23. Total fuel per hour (no reserve), pounds | 4,980 | 3,360 | 3,080 | 1,220 | 1,280 | 1,660 | 3,090 |
| 24. Total weight of machinery and boilers (wet), tons | 710 | 515 | 400 | 755 | 585 | 625 | 422 |
| 25. Machinery weight (lbs. per propeller S.H.P. or per I.H.P. per hour) | 525 | 380 | 300 | 580 | 450 | 470 | 315[11] |
| 26. Calculated fuel for 4,000 miles+20% reserve, tons[8] | 1,000 | 675 | 625 | 235 | 247 | 317 | 625 |
| 27. Feed water, tons (2% make up feed carried) | 140 | 135 | 120 | 0 | 0 | 0 | 120 |
| 28. Consumable stores, crew and effects, tons | 50 | 50 | 50 | 50 | 50 | 50 | 50 |
| 29. Gross deadweight, tons | 9,690 | 9,885 | 10,000 | 9,645 | 9,815 | 9,775 | 9,978 |
| 30. Cargo deadweight tons | 8,500 | 9,025 | 9,205 | 9,360 | 9,518 | 9,408 | 9,183 |
| 31. Length of engine and boiler rooms[9] | 91[10] | 63 | 50 | 55 | 48 | 56 | 47 |

*Notes on Table*

[1] All propellers are 3 bladed in this comparison.

[2] Propeller efficiencies from Taylor's charts; mean width ratio = .25. Wake for single screw ships = .31; for twin screw = .23.

[3] Hull efficiency for single screw ships = 1.05; for twin screw = 1.00.

[4] Efficiency ratio of S.S. "Rampo" = .58 without superheat; .59 is used in above calculations due to reduced condensation with 60° superheat.

The S.H.P. of geared turbine installation is measured beyond the gears; hence efficiency ratio includes loss in gears. For this reason efficiency ratio of turbine driving generator has been taken as .67 against .63 for geared turbine. Efficiency of gears taken as 95%; electric transmission as 92%; 1% loss allowed for geared turbine due to windage in astern blading.

[5] Steam and fuel consumptions are expressed in lbs. per S.H.P. per hour in order to make all the figures comparable. The mechanical efficiency of the reciprocating engines has been taken as 92% and all consumptions per I.H.P. have been divided by .92 before recording them in the table. The reciprocating engines are fitted with attached air pumps and Kingsbury thrust blocks.

[6] 20 lbs. of coal per sq.ft. G.S. per hour; H.S. ÷ G.S. = 36.5.

[7] Boiler efficiencies taken as 90% of test values.

[8] Port fuel oil allowed as follows: Steamship = 7 tons per day; motorship = 1 ton per day. Days in port per single voyage assumed as 7 days. Coal in port = 10 ton per day.

[9] Where there is an offset in the aft bulkhead the length of a rectangular compartment of equivalent floor area is given in table.

[10] Cross coal bunker included in this length.

[11] With induction motor, weight = 325 lbs./S.H.P.

Calorific value of coal = 14,400 b.t.u. per lb.; oil = 18,800 b.t.u.

# CHAPTER XV

## CONDENSERS

**164. General.**—The object of the condenser is to reduce the back pressure on the prime mover, thus lowering the final pressure and causing less heat to be discharged in the exhaust. The Rankine efficiency $(H_1-H_2/H_1-q_2)$ is improved and an actual reduction in steam consumption results. The increased expansion of the steam thus brought about causes more heat to be converted into useful work and a greater horsepower obtained. Also, the saving of the condensed steam by a surface condenser is, of course, absolutely necessary on shipboard.

Theoretically, the greater the vacuum the greater will be the efficiency of the prime mover; but practical considerations impose limits to which the vacuum can be economically carried. With a reciprocating engine, 25 to 27 in. is about the limit. Vacua greater than this cause an undue size for the low pressure cylinder in order to handle the increased steam volume. The increase in the capital charges due to the larger cylinders, valves, air pump and condenser, together with the losses brought about by increased cylinder condensation and friction of the engine, will more than offset the gain in economy and horsepower due to the higher vacuum.

With a turbine, the disadvantages mentioned above for reciprocating engines do not exist and the economical limit of vacuum is probably close to 29 in. The capital charges on equipment and the cost of producing vacua higher than 29 in. will, as a rule, more than offset any reduction in steam consumption. High vacua have the further disadvantage in that the whole system must be kept in the highest state of efficiency in order to keep air leakage within a reasonable figure.

The following example shows the theoretical steam consumptions for two turbines, both having the same initial conditions but one operating with 27 1/2 in. and the other 29 in. vacuum.

*Case I:*

$P_1$ (at throttle) $=275$ lbs. abs. 100°F. superheat.

$P_2$ (in condenser) $=29$ in.

$H_1-H_2=412$.

Theoretical steam consumption = 6.2 lbs. per S.H.P. per hour.

*Case II:*

$P_1=275$ lbs. abs. 100°F. superheat.

$P_2$ (in condenser) $=27\ 1/2$ in.

$H_1-H_2=370$.

Theoretical steam consumption = 6.9 lbs. per S.H.P. per hour.

Increase in steam consumption of Case II over Case I = 11.3 per cent.

**165. Air Leakage.**—The most important factor in producing and maintaining a high vacuum is the presence of air in the condenser. The main duty of the air pump is to remove the air from the condenser at the same rate at which it enters, and the cost of producing a high vacuum depends directly on the amount of air leakage. The size of the air pump and the power required to operate it depend more on the amount of air leakage than on the vacuum carried or the size of the plant. The possibility of air leakage of course increases with the amount of vacuum, the size of the propelling unit, and the amount of feed water used.

Besides increasing the size and horsepower required for the air pump, the presence of air in a condenser decreases the rate of heat transfer between the exhaust steam and the cooling water and thus lowers the efficiency of the condenser. Further, the presence of air in a condenser lowers the temperature in the condenser for a given vacuum which causes the condensate to leave the condenser at a lower temperature. This results in a lower temperature of the feed water entering the feed water heater ($q_2$) and may cause a reduced overall efficiency of the plant.

Not only must the condenser and exhaust trunk to the condenser be kept tight, but stuffing boxes and glands must be designed so there is practically no air leakage and the boiler feed water should be as free of air as possible. A large area should be given to the feed water tank to facilitate the liberation of air. Drains and auxiliary exhaust should never be led to

the condenser. An air chamber fitted on the main feed line between feed pump and the boiler will collect some of the air in the feed water. This chamber should be fitted with a vent for removing the air from time to time.

With reciprocating engines there may be considerable leakage around the L.P. valve and piston rod, as the pressure within the cylinder is less than that of the air outside. The "steam seal" adopted in the later ships of the U. S. Navy that were fitted with reciprocating engines was equipped with this device. The "steam seal" also prevents the entrance of oil on the L.P. rod into the cylinders.

In order to eliminate air leakage around the shaft into the low pressure stages of turbines, it is customary to fit labyrinth packing under steam pressure. This causes a small leakage of steam outward into the engine room instead of an entrance of air from the engine room into the turbine casing. The Westinghouse Company now fits, in addition to the labyrinth packing, a water sealed gland for use at high speeds. This gland consists of a runner similar to that of a centrifugal pump which revolves in the gland casing. The runner is supplied with fresh water and a solid annulus of water is maintained at the outer edge, thus making a seal against air leakage.

When a condenser is in operation the weight of air entering and leaving in a given time must be the same. If the volume of this air when it reaches the air pump suction is large and the air pump is not of proper size to handle it, the vacuum falls until an equilibrium has been established between the air pump and condenser. It should be borne in mind that the function of the condenser is to create the vacuum by condensing the steam and the function of the air pump is to remove the condensed water and air so that the condenser can perform its proper function at the desired back pressure.

The amount of air to be expected in a condenser can be estimated from the following authorities:

1. Orrok (A. S. M. E., 1910) gives the following figures from experiments on turbine units of 5,000 to 20,000 kw.:

1 cu. ft. free air per min. (4.5 lbs. per hour) under best conditions.
15–20 cu. ft. free air per min. (68–91 lbs. per hour) ordinary leakage.
30–50 cu. ft. free air per min. (136–230 lbs. per hour) bad leakage.

2. Hodgkinson (Trans. Soc. N. A. & M. E., 1918) states that

3 1/2 cu. ft. of free air per minute is considered good practice with a 40,000 hp. turbine unit, and 7 cu. ft. is excessive leakage.

3. Kothny (Sterling's Marine Engineers' Handbook) gives curves showing the air leakage for both turbines and reciprocating engines. These data show the air leakage increasing with the steam consumption which, as already pointed out, is often the case. For a 5,000 hp. turbine the leakage from these curves would be 20 lbs. of free air per hour (4.5 cu. ft. per minute). The data give values consistent with the figures quoted above.

In large power plants on shore air bells are often fitted to the discharge of the air pump to measure the amount of air leakage. While this practice is almost impossible for a ship operating at sea there is no reason why a test by this method cannot be used in port.

Air leakage with reciprocating engines is much greater than with turbines because of the larger number of sliding joints, relief valves, drains, etc. However, if the steam seal mentioned above is used on the L.P. cylinder this leakage can be greatly reduced over the present practice. It should also be borne in mind that reciprocating engines never carry vacua over 27 or possibly 27 1/2 in., so that possibilities of leakage and the bad effects of air in the condenser are not so great as with turbines where higher vacua are carried. Air leakage of about double those given above for turbine installations should be allowed with reciprocating engines.

**166. Thermodynamics of the Condenser.**—For a complete knowledge of the behavior of air and steam in a condenser it is necessary to have an understanding of Dalton's Law.

Dalton's Law states:[1]

1. The pressure of a gaseous mixture (composed of two or more gases or a gas and a vapor) contained within a vessel is the sum of the pressures that the separate gases would exert if each occupied the vessel alone.

2. Each constituent behaves as if the others were not present, that is, each occupies the volume of the vessel and each exerts its own pressure, corresponding to the temperature within the vessel.

[1]The student is referred to textbooks on thermodynamics for a complete statement and explanation of Dalton's Law.

A mixture of water vapor, steam, and air in a condenser follows this law closely. The temperature at any point within the condenser must be the same for both air and steam and must correspond to that given in the steam tables for the partial pressure of the steam. The vacuum or absolute pressure in the condenser is the sum of the partial pressure of the air and the partial pressure of the steam.

The following equations express Dalton's Law and give the relations of volume, pressure, and temperature in a condenser:

If $V_a$ = volume of air present.
$V_s$ = volume of steam present.
$t_a$ = temperature of air.
$t_s$ = temperature of steam.
$P_a$ = pressure of air.
$P_s$ = pressure of steam.
$W_a$ = weight of air present in condenser.

We have:

$$V_a = V_s$$
$$t_a = t_s$$
$$P_t = P_a + P_s$$
$$P_a = \frac{W_a R(t_a + 460)}{144\ V_a}$$

The following example will make clear the application of Dalton's Law to a condenser:

Vacuum in condenser = 29 in.
$P_t$ = 1.0 in. Hg. (.489 lb./sq. in.)

Suppose that the temperature existing in the lower part of the condenser is 70°F. The partial pressure of the steam ($P_s$) corresponding to a temperature of 70° is .739 in. Hg. (.363 lb./sq. in.).

We now have:

$P_t$ = .489 lb./sq. in.
$t_s = t_a$ = 70°F.
$P_s$ = .363 lb./sq. in.
$V_s$ (vol. occupied by 1 lb. steam at .739 in. Hg.) = 871 cu. ft.

$P_a = P_t - P_s = (.489 - .363) = .126$ lb./sq. in.

$$P_a = \frac{W_a R T_a}{144 V_a} = \frac{W_a \times 53.3 \times 530}{144 \times 871} = 0.126$$

$W_a = .558$ lb. of air per lb. of steam. $(P_s/P_t = .74)$

This shows that at this particular place in the condenser the ratio of air to steam by weight is .558:1.00.

The ratio of the weight of air to the weight of steam in a condenser varies through the condenser. At the inlet of the condenser the ratio of air to steam is very small and its effect is negligible. As the mixture passes through the condenser the steam is condensed and the weight of water vapor present is constantly getting smaller while the weight of air remains the same. Thus the ratio of air to water vapor increases rapidly until at the air pump suction it is very high.

The partial pressure of the air increases as the steam is condensed, becoming a maximum in the bottom of the condenser. The increase in the partial pressure of the air causes a reduction in the partial pressure of the steam, the total pressure in the condenser remaining constant. This reduction in the partial pressure of the steam results in a lowering of the temperature of the mixture in the lower part of the condenser. As a result of this lowering of the temperature in the lower part of the condenser, the temperature of the condensate is reduced, giving a lower value of $q_2$ for the water entering the feed tank. A low temperature of the condensate leaving the condenser ("hot well temperature") generally indicates a large air leakage although it may be caused by too great a quantity or the wrong distribution of the cooling water. The hot well temperature ordinarily runs from 3 to 20° lower than the temperature corresponding to the vacuum.

The total pressure (vacuum) in a condenser is generally nearly constant from the top to the bottom of the condenser although there may be a small pressure drop of 0.05 to 0.10 in. of mercury. A drop in pressure larger than this indicates that the condenser is poorly designed. The temperature in the condenser, however, gradually decreases from the top to the bottom. As already pointed out the temperature is fixed by the partial pressure of the steam which decreases as the partial pressure of the air increases.

An illustration of the changes in temperature that might exist in a condenser for a 29 in. vacuum is shown in the following table:

| Proportion of air and steam by weight | Temperature, F. |
|---|---|
| Saturated steam | 79° (at condenser entrance) |
| Air .25 steam 1.00 | 75° |
| Air .50 steam 1.00 | 71° |
| Air .75 steam 1.00 | 67.5° |
| Air 1.00 steam 1.00 | 64.5° (at air pump suction) |

The table on page 237 from a paper by Stuart and Senner in the *Jour. A.S.N.E.* 1920 (modified from a paper by Gibson and Bancel, A.S.M.E. 1915) shows the action of a mixture of one pound of air and steam in a condenser.

In the table on page 237 the air leakage has been taken as .0002 lb. of air per pound of steam. This corresponds to 1.2 cu. ft. of free air per minute with a steam consumption of 27,000 lbs. per hour.

1.2 cu. ft. free air per min. $= 1.2 \times .076 \times 60 = 5.45$ lbs. per hour.

$\dfrac{5.45}{27{,}000} = .0002$ lb. of air per pound of steam entering condenser.

A study of this table will indicate the action taking place in the condenser. At the entrance of condenser the total pressure is 1 in Hg. The partial pressure of the air by Dalton's Law is:

$$P_a = \frac{W_a RT}{V_a} = \frac{.0002 \times 53.3 \times 539}{657 \times 144} = \frac{1}{16{,}000} \text{ lb. per sq. in.}$$

$$= \frac{1}{8{,}000} \text{ in. Hg.}$$

where $V_a = V_s$ corresponding to the volume of steam at 79° which is the temperature at the top of condenser.

$W_a =$ pounds of air present per pound of steam.

As the steam is condensed by the cooling water, the weight of steam decreases rapidly from the top to the bottom of the condenser. The above equation will show that the partial pressure of the air ($P_a$) increases as the volume of vapor is decreased. (Line 3.)

The increase in the partial pressure of the air causes a decrease

TABLE XIII

**Table** showing typical action in a **condenser** supplied with 1 lb. of **steam-air mixture**, **containing 0.0002 lb. air**. (**1.2 cu. ft.** per min. with **27,000** lbs. steam per hour.) **Illustrating the process of steam condensation and air compression at a constant vacuum** of 29 inches.

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 1. Temperature of mixture, deg. F.... | 79 | 79 | 79 | 79 | 76 | 72 | 59 |
| 2. Partial pressure of steam, in. hg.... | 1.0 | 1.0 | 1.0 | 0.99 | 0.90 | 0.80 | .50 |
| 3. Partial pressure of air, in. hg....... | 1/8,000 | 1/4,000 | 1/1,000 | 1/100 | 1/10 | 1/5 | 1/2 |
| 4. Pounds of air present............. | .0002 | .0002 | .0002 | .0002 | .0002 | .0002 | .002 |
| 5. Pounds of steam condensed........ | 000 | .5000 | .8748 | .9873 | .9987 | .9993 | .99967 |
| 6. Pounds of steam present.......... | .9998 | .4998 | .1250 | .0125 | .0011 | .0005 | .00013 |
| 7. Volume of air present, cu. ft........ | 657 | 328 | 82 | 8.2 | .823 | .412 | .164 |
| 8. Volume of mixture, cu. ft.......... | 657 | 328 | 82 | 8.2 | .823 | .412 | .162 |
| 9. Volume of mixture, cu. ft. per lb. air | 3,285,000 | 1,640,000 | 410,000 | 41,000 | 4,115 | 2,060 | 820 |
| 10. Lbs. of air per 1,000 cu. ft. mixture. | .0003 | .0006 | .0024 | .024 | .243 | .485 | 1.22 |
| NOTE: | Condenser entrance | | | | | | Air pump suction |
| | Steam condensing in this zone. | | | | Air cooling and compressing in this zone. | | |

in the partial pressure of the steam (Line 2) since the total pressure is constant. This decrease in $P_s$ causes the temperature in the condenser to decrease (Line 1) $t_s = t_a =$ temperature corresponding to $P_s$ in the steam tables.

Both the increase in the partial pressure of the air and the decrease in the condenser temperature from top to bottom cause the volume occupied by the given weight of air to be decreased. (Line 7.)[1]

A decrease in the volume of the air is, of course, necessary in order to reduce its volume so that it can be handled by the air pump. Thus the bottom of the condenser serves the purpose of reducing the volume of the air although, as already pointed out, it results in the lowering of the temperature of the condensate. Later on, under the discussion of air pumps, this will receive further treatment. The greater the air leakage per pound of steam the greater will be the depression of the hot well temperature. If there is a fall in the total pressure in the condenser from top to bottom, as is often the case, the temperature will be further depressed.

**167. Heat Transfer.**—Orrok, who has published by far the most complete and reliable information on heat transfer in condensers, gives the following law of heat transfer between steam and water:

$$U = 350\, C_1 C_2\, (P_s/P_t)^2 \sqrt{V}$$

where $U$ = b.t.u. transfer per sq. ft. per degree difference per hour.

$C_1$ = cleanliness coef. (varies between 1.00 and .50).
$C_2$ = material coef. (see table).
$V$ = velocity of cooling water in ft. per sec.
$P_s$ = partial pressure of steam in condenser.
$P_t$ = total pressure of steam and air.

The air richness factor $(P_s/P_t)$ will of course vary throughout the condenser, increasing rapidly at the bottom. Orrok (Mark's Handbook, p. 1009) gives values of $(P_s/P_t) = 0.95 - 0.97$ for tight condensers. Table XIII shows this ratio varying between 1.00

[1] At 59°, $P_a = (.488 - .247) = .241$ lbs/sq. in. The volume occupied by the air that entered with one pound of steam (.0002 lbs.) is,

$$V_a = \frac{.0002 \times 53.3 \times 519}{.241 \times 144} = 0.16 \text{ cu.ft.}$$

and .80 for a temperature drop of 7°; and 1.00 to .50 for a temperature drop of 20°. Probably the first figures represent the case for the larger part of the condenser as nearly all the heat transfer in condensing takes place in the upper part of the condenser. Experiments with two pass condensers show that about 3/4 of the heat transfer between the steam and cooling water takes place in the second (upper) pass of the circulating water. A mean of 1.00 and .80 gives $P_s/P_t = .90$, which no doubt is too low a value for the major part of the condensing surface. With a two pass condenser this air richness ratio would be nearly unity in the upper pass and .70 to .90 in the lower pass. Therefore, values in the neighborhood of .95 would appear reasonable for ordinary leakage. The foregoing brings out clearly the effect that the presence of air has on the rate of heat transfer.

Table XIV

*Material coefficient* ($C_2$)

| | | | |
|---|---|---|---|
| Admiralty | .98 | Zinc | .75 |
| Copper | 1.00 | Monel metal | .74 |
| Aluminum lined | .97 | Shelby steel | .63 |
| Admiralty (oxidized) | .92 | Admiralty (badly corroded) | .55 |
| Aluminum bronze | .87 | Admiralty (vulcanized inside) | .47 |
| Cupro-nickel | .80 | Glass | .25 |
| Tin | .79 | Admiralty (vulcanized both sides) | .17 |
| Admiralty (lead lined) | .79 | | |

A study of the formula for heat transfer given above shows that the rate of heat transfer varies as the square root of the velocity of the cooling water. Orrok recommends a velocity of the cooling water ($V$) of 8 ft. per second for high vacua. If a higher velocity of cooling water is used a smaller and lighter condenser can be obtained by reducing the cooling surface. In high speed craft this, of course, is of importance. Further, a high water velocity reduces the possibility of incrustation and scale on the condenser tubes. Velocities as high as 15 ft. per second are used in high speed ships where weight and space are important.

With low vacua condensers, where the rise in the temperature of the cooling water between inlet and discharge is large, due to the higher temperature existing in the condenser, the speed of the cooling water will have to be reduced below 8 ft. per second in order to keep the length of the condenser within reasonable

limits. Velocities of 3 to 5 ft. per second should be used with condensers where the vacuum is in the neighborhood of 24 to 27 in., unless a three pass condenser is used.

The foregoing is clearly illustrated in Art. **198,** and where computations for condenser design are carried through.

The horsepower of the circulating pump varies as the cube of the velocity of the cooling water ($V_3$); so high velocities of

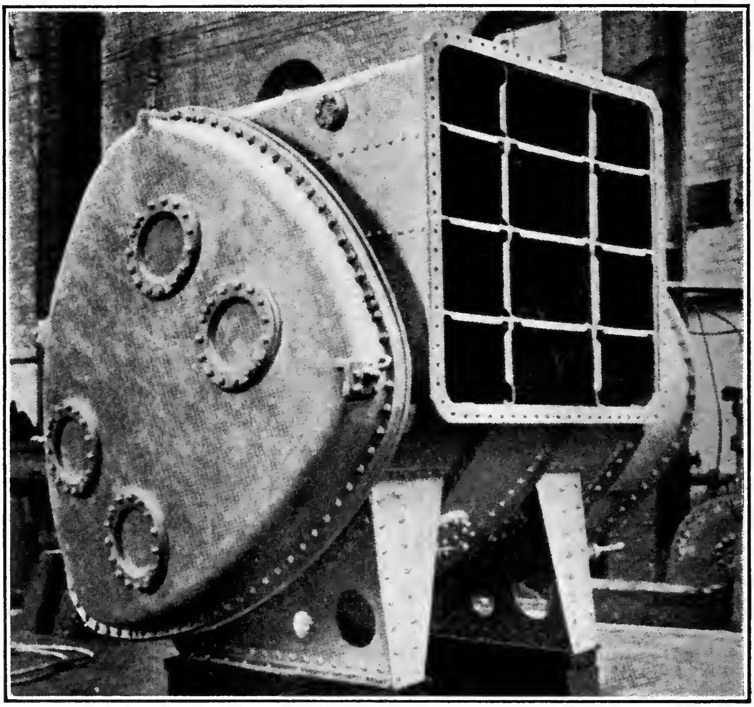


FIG. 87.—Weir "Uniflux" condenser.

the circulating water, while reducing the size of the condenser, cost more to maintain.

A full explanation and discussion of the methods of obtaining the amount of circulating water and cooling surface is given under condenser design in Chap. XIX.

**168. Types of Condensers.**—There are a great many types and shapes of condensers—some with baffles and some with special tube spacing. Probably one of the best and simplest types is the pear shaped condenser with regular tube spacing

and without baffles similar to that used by Weir, Fig. 87. Many claims are put forward for special shapes and special interior arrangements of tubes and baffles, but the simple type of pear

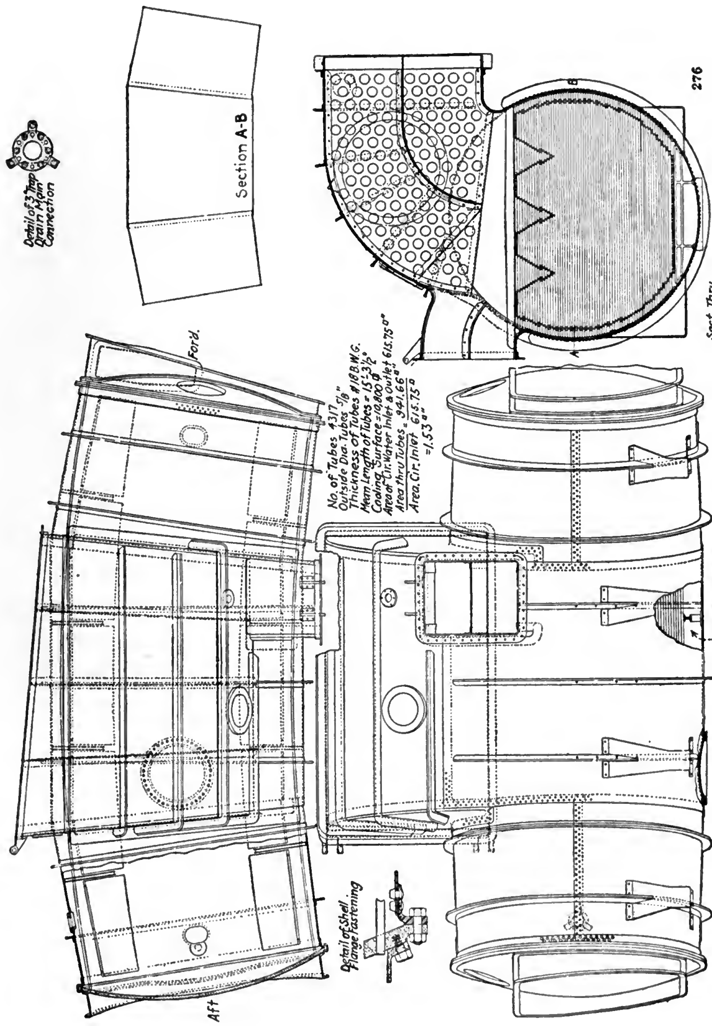



FIG. 88.—Destroyer condenser with tubes expanded in both tube sheets.

shaped condenser has given good, consistent results in practice. In this shape of condenser the area decreases as the steam is condensed, thus keeping the same high velocity of the steam from top to bottom of condenser.

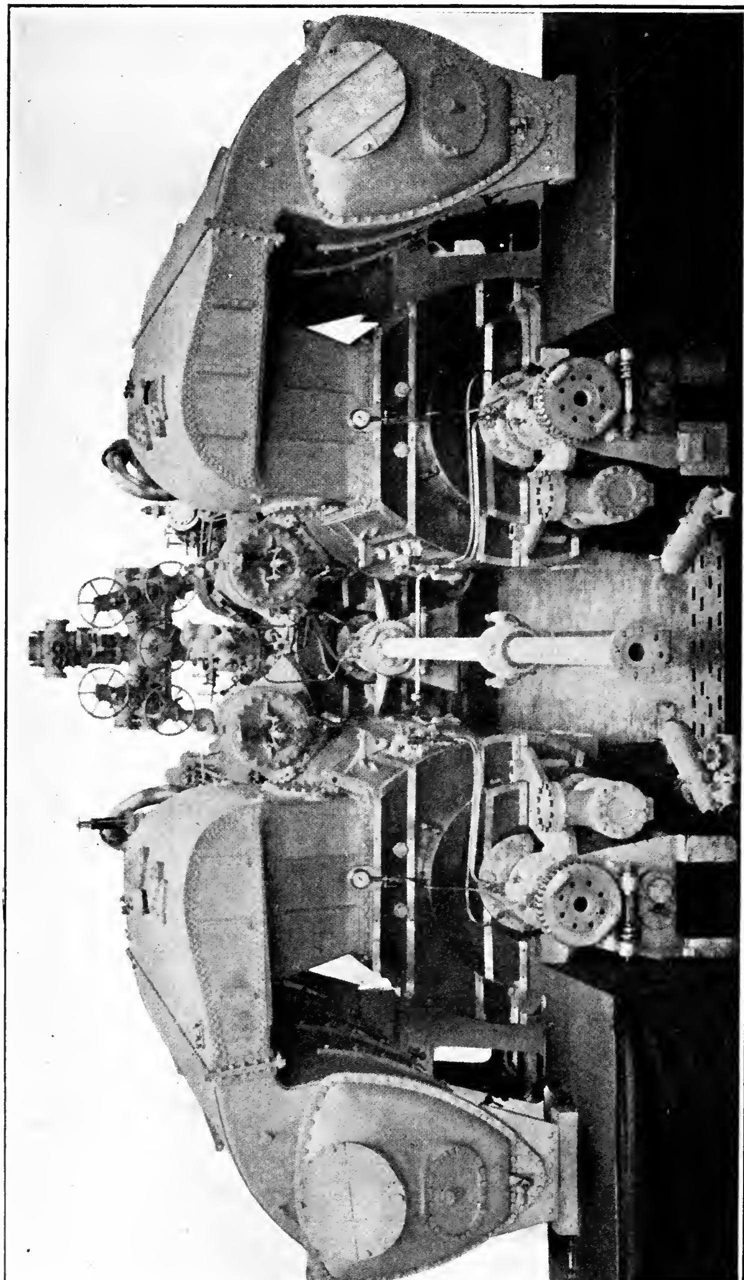


Fig. 89.—Weir condensers on turbine-driven destroyer.

There is no doubt that the circular shape is not so efficient as the pear shape, but it makes a cheap and strong construction and for this reason is often used (Fig. 88).

The air should always be withdrawn from the lowest part of the condenser where its volume has been reduced to a minimum. For high vacua installations separate suctions should be provided for the condensate and the air, the air being withdrawn above the water surface a small distance above the bottom. Too often,

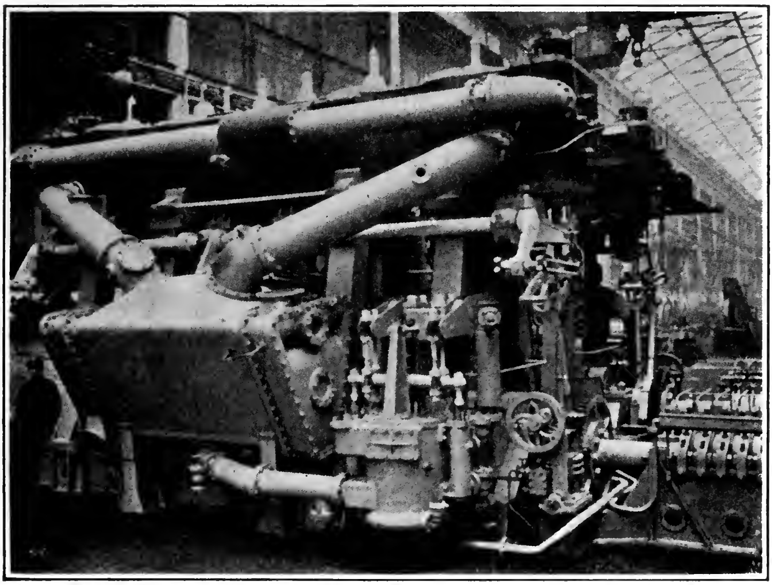


Fig. 90.—Engine-driven air pump and Weir condenser. Reciprocating engine installation.

even with high vacuum, only one suction is provided at the lowest point as shown in Fig. 97. This is the universal practice with reciprocating engines where only one air pump is provided for both condensate and air.

The condenser should be located as close to the prime mover as possible in order to reduce the possibility of air leakage in the exhaust nozzle and to reduce the pressure drop between the engine and condenser.

The exhaust trunk from the turbine or engine should be designed to discharge along the full length of the condenser. This

is the universal practice with turbine installations as shown in Fig. 89 but is not so general with reciprocating engines (see Fig. 90).

Practically all condensers are of the two pass type. The first pass of the circulating water should be through the lower part of the condenser. This gives a large temperature difference between the steam and the cooling water in the lower part of the condenser, thus keeping the volume of air to be handled by the air pump at a minimum. This arrangement has the further advantage in giving a larger temperature difference in the lower part where $(P_s/P_t)$ is small and the coefficient of heat transfer $(U)$ consequently small. Three pass condensers are used where space is limited either for the condenser or for withdrawing tubes, and in low vacua installations to reduce the length of the condenser. Three pass condensers are objectionable because of the added turbulence of the water, the baffles required at each end and because of the fact that the circulating water leaves at the opposite end from which it enters.

As mentioned in Chap. IV it is very essential that all salt water be kept out of the feed water. This is especially so when water-tube boilers are used. Practically the only place that salt water can leak into the feed water system is in the condenser where the salt circulating water is passing through the condenser tubes.

The flat tube sheets at the end of the condenser require staying on account of the difference of pressure between the inside and outside of the condenser and leakage is liable to take place where the stays are secured to the tube sheets. The tubes are generally held in the tube sheets by screwed ferrules and corset lacing packing and these tube packings are also a source of salt water leakage. The practice in recent years in the U. S. Navy has been to use tubes expanded into the tube sheets. This does away with the troublesome packing but requires a curved tube to take care of expansion. A type of condenser with tubes expanded in both ends is shown in Fig. 88. This has been taken from a paper by Dinger in the Journal of the A. S. N. E., May, 1914. The condenser has been curved to allow for expansion and the stays have been eliminated as the expanded tubes serve to stiffen the ends. It is claimed that salt leaks have been practically eliminated with this type of condenser.

The latest practice is to pack condenser tubes with metallic packing similar to that used in packing piston rods. A type of spirally wrapped babbitt foil packing, manufactured by the Crane Packing Co., has recently been used for packing condenser tubes with excellent success and is being widely adopted both in the merchant and naval service.

The following table shows the performance of some "Weir-Uniflux" Condensers:

| Installation | Turbine torpedo boat | Turbine vessel X | Turbine vessel Y | Turbine vessel Y | Turbine vessel Y |
|---|---|---|---|---|---|
| Steam condensed, lbs./hr. | 370,000 | 34,200 | 29,000 | 37,000 | 46,000 |
| Surface, sq. ft. | 12,500 | 1,800 | 1,300 | 1,300 | 1,300 |
| Condensation rate, lbs. sq. ft. | 30 | 19 | 22.4 | 28.6 | 35.4 |
| Vacuum at 30 in. barometer | 28.1 | 28.9 | 28.36 | 28 | 27.3 |
| Circulation inlet temp., deg. F. | 54 | 56 | 54 | 54 | 54 |
| Circulation outlet temp., deg. F. | 76 | ........ | 74 | 78 | 86 |
| Hot well temperature, deg. F. | 83 | 76 | 84 | 88 | 101 |
| Type of air pump | Monotype | "Dual" | | | |

**169. Lovekin Condenser.**—Among the various makes of condensers that have adopted special shapes and special tube spacing to improve the rate of heat transfer the Lovekin condenser (Fig. 91) is of special interest. In this condenser the depth of the tube bank has been reduced to about half that ordinarily used and passages have allowed leading to the bottom of the condenser for the condensate to pass through. A separate bank of tubes in the first pass of the cooling water is fitted in the lower part of the condenser so constructed that the condensate from the upper part of the condenser cannot drip over them. This bank of tubes acts as an air cooler and is connected, as shown in the cut, directly to a "Radojet" air ejector.

The advantage of this condenser is evident. The upper tubes serve for condensing the steam and the lower tubes serve for cooling the air the same as in all condensers. By the special arrangement adopted the condensate is not cooled by passing over the lower tubes as is the general practice, but passes directly to the hot well chamber in the bottom of the condenser, where it is drawn off by the condensate pump at a temperature very close to that corresponding to the vacuum. Thus the con-

denser cools the air and reduces its volume so that it can be handled by the air pump; yet the condensate is withdrawn at a temperature from 8° to 20 higher than with many types of

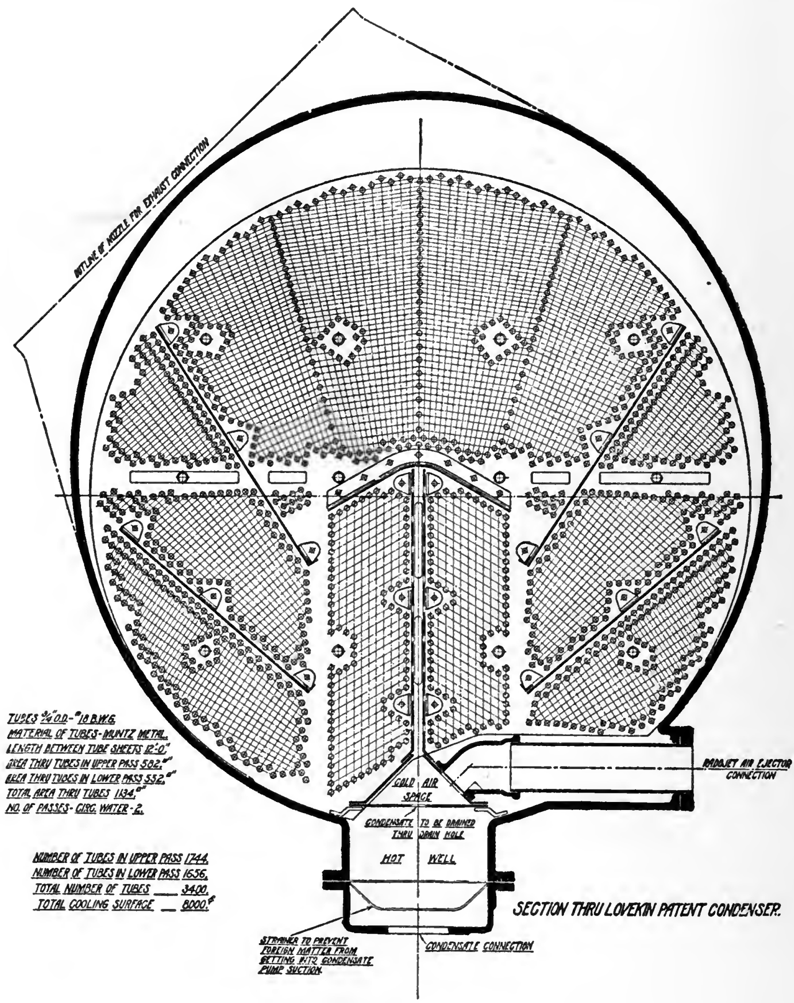



FIG. 91.—Lovekin condenser.

condensers which, of course, improves the overall plant efficiency.

A description of this condenser is given by Mr. Lovekin in a paper before the Society of Naval Architects and Marine Engineers, 1920. The following tests on a condenser of this

type are taken from this paper. Attention is especially directed to Line 9, showing the temperature difference between the steam in the condenser and the cooling water exit.

DATA ON CONDENSER SHIP "CANTIGNY"

| No. | Item | Unit | | |
|---|---|---|---|---|
| 1 | Run No. | | 1 | 2 |
| 2 | Barometer | in. hg. | 30.08 | 30.03 |
| 3 | Avg. vacuum in shell | in. hg. | 28.94 | 28.93 |
| 4 | Absolute press. in shell | in .hg. | 1.14 | 1.10 |
| 5 | Corresponding temp. $t_s$ | Deg. F. | 83.2 | 82.1 |
| 6 | Inlet water temp. $t_1$ | Deg. F. | 58.3 | 58.7 |
| 7 | Discharge water temp. $t_2$ | Deg. F. | 76.6 | 77.2 |
| 8 | $t_s - t_1$. | Deg. F. | 24.9 | 23.4 |
| 9 | $t_s - t_2$ | Deg. F. | 6.6 | 4.9 |
| 10 | Mean temp. diff. $\frac{\text{Item 12}}{\text{Item 11}}$ | | 13.78 | 11.83 |
| 11 | Surface, outside | sq. ft. | 8,011 | 8,011 |
| 12 | Steam condensed | lb./hr. | 72,000 | 72,000 |
| 13 | Circulating water | lb./hr. | 3,888,200 | 3,888,200 |
| 14 | Heat per lb. steam | b.t.u. | 988 | 999 |
| 15 | $K = \frac{\text{B.t.u. per hour}}{\text{sq. ft.} \times \text{deg.}} = \frac{(17) \times (15)}{TD\ (14) \times (13)}$ | | 645 | 759 |

Velocity of circulating water through tubes = 4.37 ft. per second.

**170. Auxiliary Condenser and Deck Winches.**—One of the functions of the auxiliary condenser is to take care of the surplus of the exhaust steam from the auxiliaries which is not needed for feed heating. As the heat of vaporization of this exhaust steam discharged to the auxiliary condenser is carried away by the cooling water, it is obvious that the amount of steam flowing to this condenser should be a very small amount. In well designed plants the auxiliary condenser is not operating under normal service conditions.

The main function of the auxiliary condenser, as installed in many ships, is to take care of the exhaust from the auxiliaries while in port. As a rule it operates at nearly atmospheric pressure, merely serving to save the condensate and not to lower the back pressure. The exhaust from the deck winches

constitutes by far the largest part of the exhaust entering this condenser in port. The heat of vaporization is carried away by the circulating water for the condensate is already at the same temperature as the exhaust and no steam is needed for feed heating.

This practice is extremely poor engineering and the adoption of electric deck winches will obviate this bad practice. Deck winches are probably the most inefficient auxiliaries on shipboard, and this is especially true in cold weather when the condensation in the deck steam lines and winch cylinders is

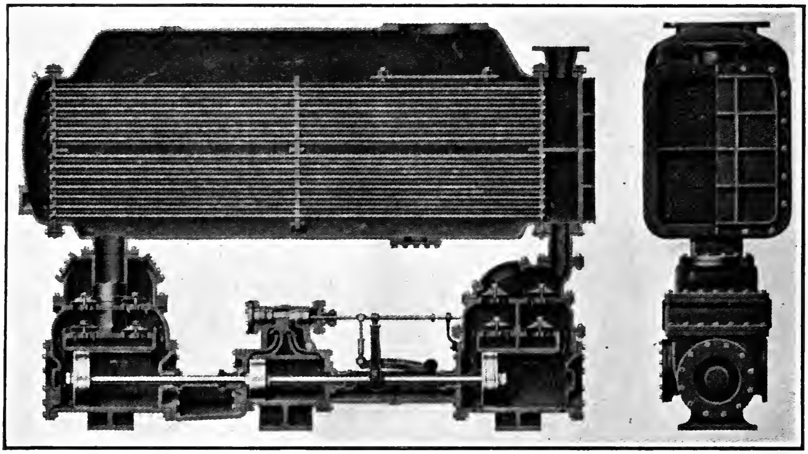


Fig. 92.—Wheeler condenser with combined air and circulating pumps.

high. Thus we have high steam consumption, heavy condensation, and a large part of the heat of the exhaust thrown away.

If electric driven winches are used the high steam consumption is eliminated. The current can be furnished by a turbine driven generator working under a high vacuum and exhausting to the auxiliary condenser. A better arrangement would be to install a Diesel generating set for port use. This will eliminate entirely the use of steam in port and allow the fires to be drawn in all boilers except when needed for heating purposes. With electric driven winches shore current may be used when available and thus allow the ship's whole plant to be shut down. This will relieve the engine force in port and allow repairs and overhauling to be carried out with great facility.

It is thus seen that the auxiliary condenser has very little useful function in the modern merchant ship. If the surplus of auxiliary exhaust not needed for feed heating is discharged to the L.P. stages of the main turbines it should be possible to eliminate it.

Auxiliary condensers when fitted have a cooling surface of roughly about 3 per cent of that the main condenser. This is an arbitrary manner of fixing the size based on previous experience. The condenser air pump and circulating pump are often a self-contained unit as shown in Fig. 92.

## CHAPTER XVI

## AIR PUMPS

**171. General.**—The function of the air pump is to remove the air and condensed steam from the condenser so that the condenser can continously produce the desired vacuum. With a low vacuum such as carried with reciprocating engines a single "wet" air pump is sufficient to take care of the condensed steam and air leakage. When a vacuum of 28 to 29 in. is carried, the air pump becomes one of the most important auxiliaries. The higher the vacuum, naturally the lower is the pressure in the condenser and the greater is the volume occupied by a given weight of air. Hence an air pump of large capacity is required at high vacua to take care of this large volume of air. As already pointed out, the amount of air to be removed depends not on the size of the plant or the amount of steam condensed but upon the amount of air leakage into the system; although the larger the installation, the greater is the tendency for air leakage.

By Dalton's Law, it will be remembered that the total pressure within the condenser is the sum of the air pressure and the steam pressure. Table XIII shows that at the entrance to the condenser the partial pressure of the air is very small, resulting in a very large volume occupied by the weight of air present—a volume altogether too large to be handled by an air pump. As the steam is condensed in its passage downward through the condenser, the volume occupied by a given weight of steam becomes smaller and smaller. Hence the ratio of air to steam by weight becomes larger and larger. This causes an increase in the partial pressure of the air and a decrease in the partial pressure of the steam. As already pointed out in the discussion of condensers, the reduction in the partial pressure of the steam results in a reduction of the temperature in the condenser. The increase in the partial pressure of the air and the decrease in the temperature cause compression of the air present in the condenser to

a much smaller volume. Table XIII shows this clearly. In this case the temperature of air pump suction has been depressed 20°, which is rather higher than necessary.

If the air leakage in a given installation should increase, the circulating pump must be speeded up and the air pump suction depressed so that the air present will be reduced in volume sufficiently to be handled by the air pump installed. If the air pump cannot handle the increased leakage, the partial pressure of the air will rise, causing the vacuum to fall until a balance is set up between the air pump and condenser.

With high vacua, as a rule, two pumps are used: a "dry air" pump handling air and water vapor and a "wet air" pump handling the condensed steam. The best types of dry air pumps are designed to reduce to the volume of the air either by decreasing its temperature as in the Weir "Dual" air pump or by compression as in the Parsons augmentor or "Radojet" ejector.

The Weir "Dual" air pump consists of two cylinders, one acting as a "dry" air pump and the other as a "wet" air pump. The "dry" air pump has an ingenious system whereby cold water is injected into the cylinder to cool the air and thus reduce its volume. The Parsons augmentor withdraws the air by a steam ejector and delivers the air to an augmentor condenser where the ejector steam is condensed. The condensate and air from both the main and augmentor condenser is handled by a single reciprocating wet air pump.

**172. Ejector Type of Air Pumps.**—The steam ejector type of air pump, of which the Wheeler "Radojet" and Westinghouse-Le Blanc are examples, is being widely used today and is giving excellent satisfaction. These use steam jets to withdraw and compress the air. The steam and entrained air are discharged into the feed tank where the heat in the steam used in operating the ejector is utilized in heating the feed water. The condensed steam is withdrawn from the condenser by a separate condensate pump.

The ejector type of air pump is much smaller, lighter and more compact than the reciprocating types. It uses more steam, however, than the latter. If this steam can be used advantageously for feed heating, as is often the case, this is not a serious objection. However, it often happens, as is shown by

the example in Art. 205, that there is more exhaust steam than is necessary for feed heating. In such a case there may be a disadvantage in using an air pump of this type. Where the ejector steam can be used advantageously for feed heating this air pump, either with or without intercondenser, is an excellent type for high vacua.

**173. Westinghouse-Le Blanc Air Ejector.**—Figure 93 shows a diagrammatic arrangement of the Westinghouse-Le Blanc air ejector system. This system has been installed on upwards of 300 merchant ships and on a number of battleships and scout cruisers.

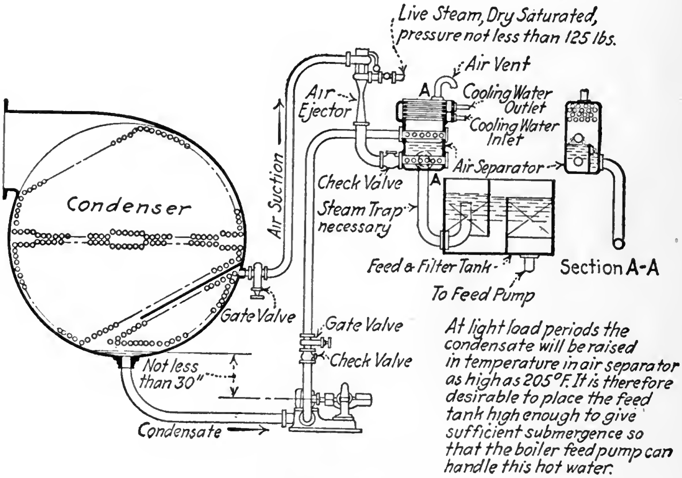



Fig. 93.—Westinghouse-Le Blanc air ejector.

The air ejector for withdrawing the air, and the condensate pump for handling the condensed steam, are clearly shown in this diagram. The air ejector consists of two sets of steam nozzles, arranged in series for withdrawing the air from the condenser. The air is entrained by the steam, compressed from the vacuum pressure to atmospheric pressure by the velocity of the steam and discharged with the exhaust steam to an air separator.

The discharge from the air ejector enters the lower part of the air separator and the discharge from the condensate pump enters the air separator higher up. Thus this piece of equipment acts as a jet condenser, condensing the exhaust from the

ejector. The air escapes into the engine room through a vent in the top and the condensate absorbs the heat in the ejector exhaust and drains to the feed and filter tank. The upper part of the air separator consists of a small surface condenser supplied with circulating water from the main circulating pump. This upper portion serves when starting up and at light loads and condenses the small portion of vapor contained in the escaping air. The heat of vaporization absorbed by the condensate in the lower part is returned to the boiler; that absorbed by the cooling water in the upper part is lost in the overboard discharge of the circulating pump.

If there is a surplus of exhaust steam available for feed heating, an intercondenser may be installed between the two stages of the ejector and the steam consumption reduced about 50 per cent.

**174. "Radojet" Air Pump.**—The following description of the operation of the "Radojet" air pump is taken from the publication of C. H. Wheeler Mfg. Co., and refers to Fig. 94:

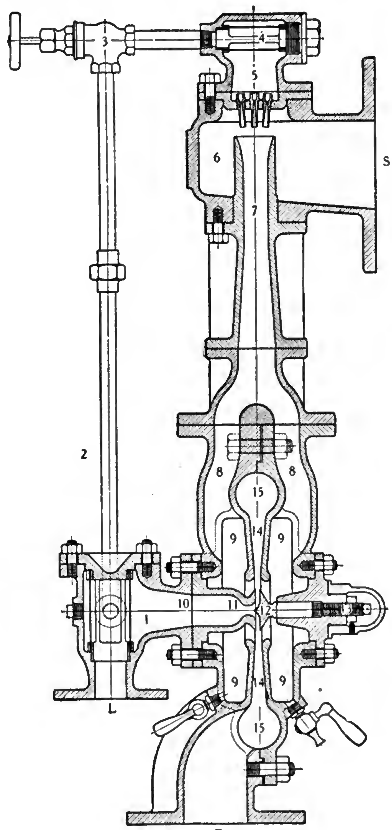


Fig. 94.—Cross-section of Radojet vacuum pump.

Live steam is delivered from a source not shown through opening *L*, through strainer cage 1, pipe 2, auxiliary steam valve 3, strainer cage 4, expansion nozzles 5, across suction chamber 6, of the first stage ejector, which is in communication with the condenser through the suction opening *S*. The steam expands in the nozzles, leaving with a very high velocity and while passing across suction chamber 6, entrains the air and vapors to be compressed. The mixture passes into the diffuser 7 from where it is dis-

charged at higher absolute pressure than that of the air entering at $S$, into a double passage 8 communicating with the suction chambers 9 of the second stage. These two suction chambers 9 are annular, giving the commingled fluid a large entrainment surface. Steam is simultaneously delivered through the strainer cage 1 into passage 10 which communicates with the annular expansion nozzle formed between nozzle 11 and nozzle point 12. Nozzle point 12 may be adjusted toward or away from disc 11 by the adjusting screw 13.

The steam delivered radially by the annular nozzle 11 expands between same and nozzle point 12, leaving it as a jet of high velocity in the form of an annular sheet, and in passing across the suction chambers 9, entrains

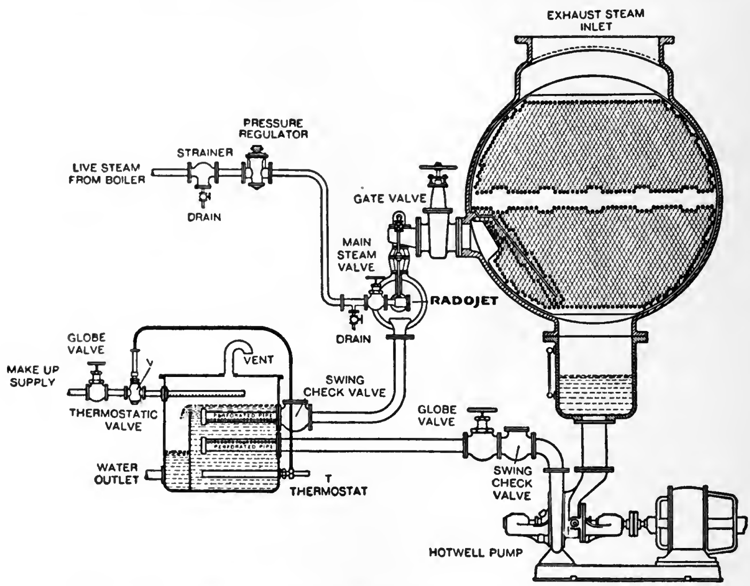



Fig. 95.—Diagram of condenser, radojet, and condensate pump.

the commingled air and steam coming from the first stage and carries them into the annular diffuser 14, thereby compressing the mixture to atmospheric pressure and discharging it into casing 15 which has the discharge opening $D$.

The mixture discharged at $D$ is delivered to the feed tank as shown in Fig. 95. Figure 96 shows this type of air pump fitted with an intercondenser between the two stages of the pump.

The operation with intercondenser is as follows:

The air to be compressed enters at $S$ into the suction chamber of the first stage, where it is acted upon in known manner by the steam issuing

in jets from nozzles. The mixture of air and steam is delivered into the chamber *B* and passes downwardly through the condenser tubes *t* around which is flowing the cooling water which condenses most of the steam. The condensate remains in chamber *C* and passes off through pipe *p*, while the air and steam continue their passage upward through the condenser tubes *t* of the second group, where the mixture is further cooled and

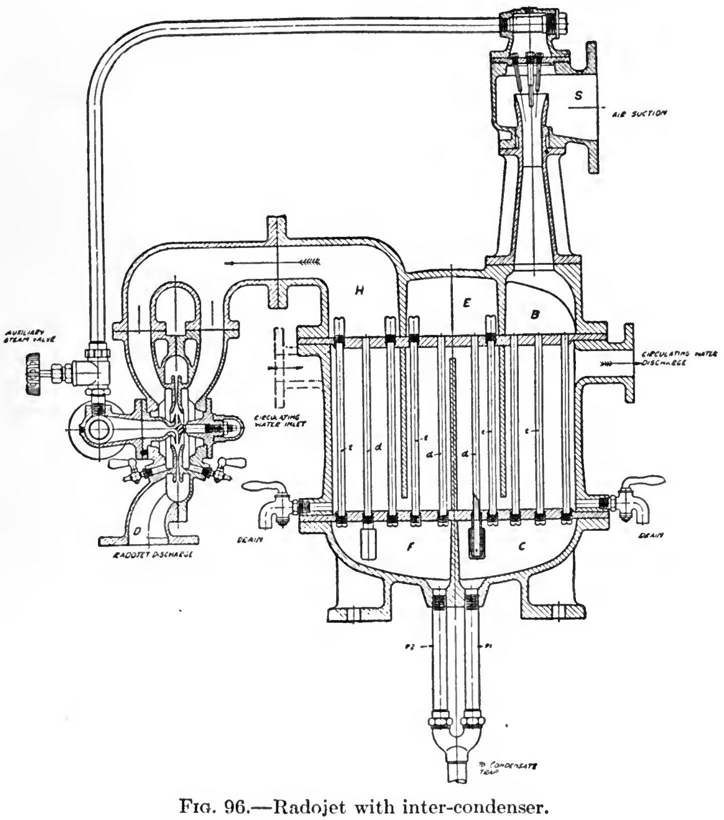



Fig. 96.—Radojet with inter-condenser.

more steam is condensed. The condensate drips downward through the drain tubes *d* into the chamber *C*. The air and remaining steam descend through the tubes *t* of the third group from chamber *E* into chamber *F*, the condensate passing off through pipe *P*. Finally the air ascends again through the fourth group of tubes *t*, any remaining condensate dripping downward and passing off through drain pipes *P*.

The air arriving in the chamber *H*, being free from steam or vapors,

passes to the suction chamber of the second stage, where it is compressed to above atmospheric pressure as previously described.

The condensate from the intercondenser is delivered through a trap to the condensate pump suction. The cooling water for the intercondenser is supplied by the main circulating pump as shown in Fig. 96.

It is claimed that by use of the intercondenser a considerable saving in steam consumption is effected. Although the intercondenser somewhat complicates the installation, it may prove a useful addition when the supply of exhaust steam is greater than required for feed heating.

The following extract of test readings made on a C. H. Wheeler surface condenser show clearly that by the use of a "Radojet" the air is effectively removed from the condenser and that the hot well temperature is close to that of the entering exhaust steam.

Size of turbine, 10,000 kw. Cooling surface, 21,000 sq. ft.

| | |
|---|---|
| Load in kw. on turbine | 11,200 |
| Barometer | 30.01 |
| Vacuum at turbine exhaust (mercury column) | 28.15 |
| Vacuum at "Radojet" suction (mercury column) | 28.25 |
| Temperature of turbine exhaust, deg. F | 98 |
| Temperature of circulating water inlet, deg. F | 75 |
| Temperature of circulating water discharge, deg. F | 90 |
| Temperature of hot well, deg. F | 94 |
| Steam consumption of "Radojet," lbs. per hr | 600 |

**175. Weir Dual Air Pump.**—The following description of the Weir air pump is taken from the publication of G. & J. Weir, Ltd., and refers to Figs. 97 and 98.

Referring to Fig. 97 this shows in a diagrammatic form the arrangement of Surface Condenser, Dual Air Pump, and Injection Water Cooler, the entire combination being of novel design and giving high vacuum and thermal efficiency. In all cases the pump *A* or wet pump is situated below the steam cylinder, as this pump is the only one which works under any considerable load; the dry pump *B* is driven by the beam and links in the usual manner. One connection *C* is made to the condenser, but a branch pipe *D* is led to the dry pump, and connection being made in such a manner that the water will all pass by *C′* to the wet pump. Both pumps are generally of the three valve marine type, but in certain cases the dry pump may be of the suction valveless type.

The first and most important difference from an ordinary twin pump consists in the separate suction to each pump, then in the dry pump dis-

charging through the return pipe $E$, through a spring loaded valve $F$, into the wet pump at a point below its head valves. The next point concerns the supply of water to the dry pump for water sealing, clearance filling, cooling, and vapor condensing. When starting the pump the filling valve $G$ must be opened for a minute or so to enable the vacuum to draw in a supply from the hot well of the wet pump. The valve is then closed, and the water passes from the hot well of the dry pump by the pipe $H$ to the annular cooler, through which a supply of cold sea water circulates, and after being cooled passes into the suction of the dry pump; then passing through the pump it becomes heated and again passes to the

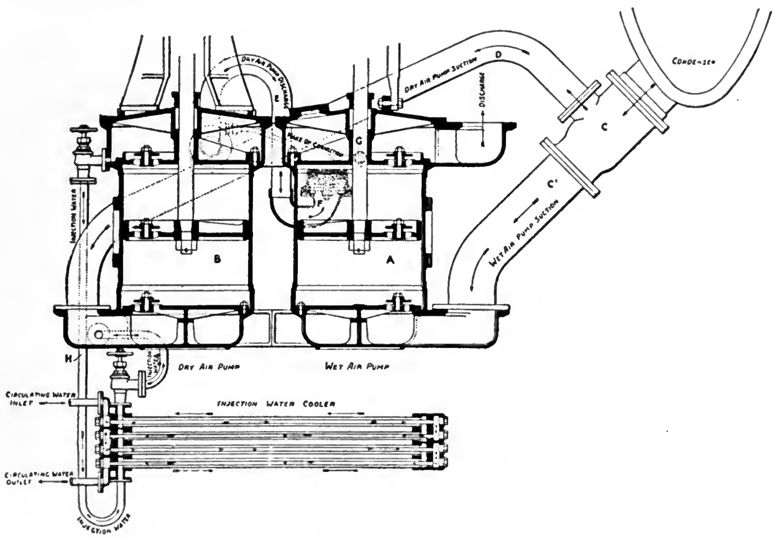



FIG. 97.—Diagram of Bethlehem-Weir dual air pump.

cooler, and so on in a continuous closed circuit, any excess passing over the pipe $E$ to the wet pump. The spring loaded valve $F$ is adjusted to maintain about 20 in. vacuum in the dry pump hot well when the condenser is working at 28 in. vacuum, and this 8 in. difference of pressure is sufficient to cause the water to overcome the cooler friction and pass into the suction, and at the same time never allow any direct air connection between the dry suction and discharge.

Attention should be directed to the method of reducing the air volume by the use of the Weir dual air pump in contrast with other types. This pump withdraws the air and condensate from the condenser at a temperature from 3 to 10° below that corresponding to the vacuum. The air is then delivered to the

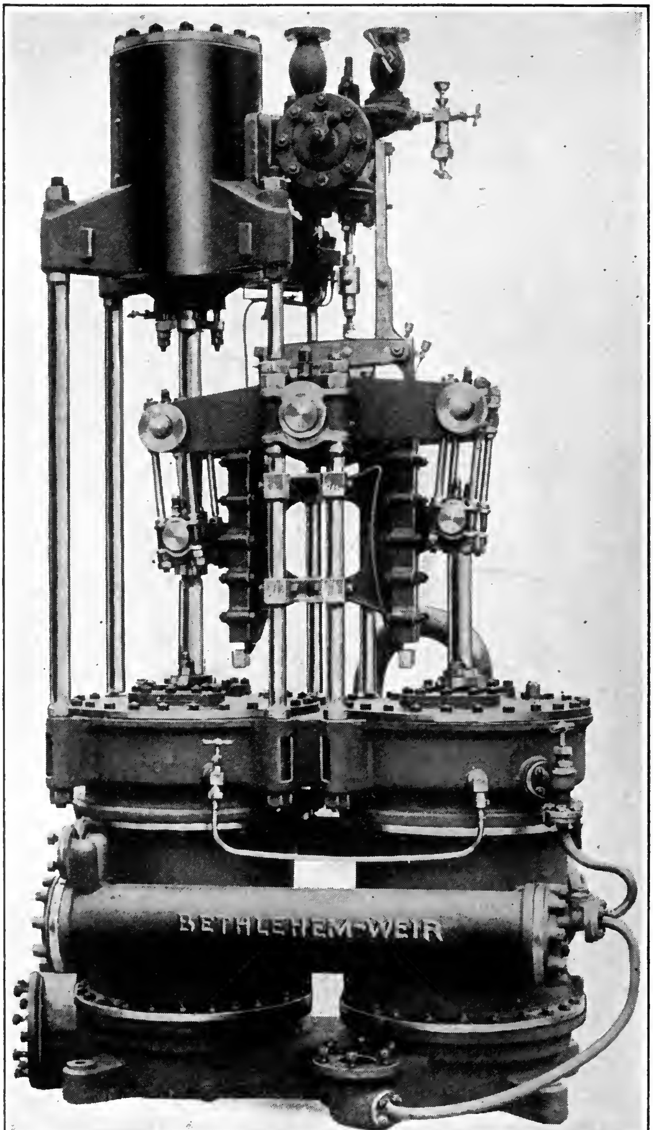



FIG. 98.—Bethlehem-Weir dual air pump.

dry air pump where its volume is reduced by cold water injection; the condensate is handled by the wet air pump at the temperature at which it was withdrawn from the condenser. Thus the air is handled without cooling the condensate and a consequent lowering of the feed temperature. On the other

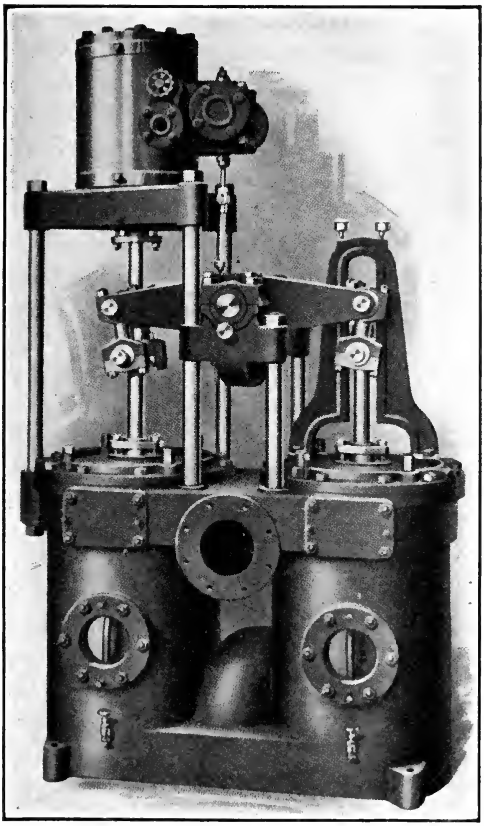


FIG. 99.—Blake-Knowles beam air pump.

hand, with other reciprocating types the air volume is reduced by cooling while it is still in the condenser. This is accomplished by lowering the "hot well" temperature (depression of temperature of air pump suction) and results in a lowering of the temperature of the condensate as well as the air.

**176. Types of Air Pumps.**—In reciprocating engine installa-

tions a single wet air pump is used for handling both the condensate and air. This is almost universally driven by a rocker arm from the main engine, Fig. 90. For an engine working at a low vacuum this arrangement has given satisfaction although for moderately high vacua of 26 to 27 inches an independently operated pump would no doubt show better efficiency. Figure 99 shows a twin air pump manufactured by the Worthington Pump & Mach. Corp. which is used to a large extent for vacua up to about 28 inches. It is very similar to the Weir pump already mentioned except it does not have the cooling attachment and other special features of the dry air pump. Other ejector air pumps are on the market that use water instead of steam as the ejecting medium; rotary air pumps are also used to some extent.

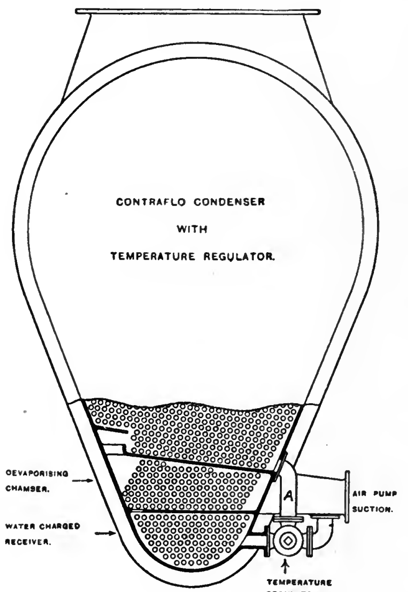



FIG. 100.

The condenser and air pump should be designed so that the condensed steam does not remain in the bottom of the condenser for any appreciable time. If water is allowed to collect in the condenser in contact with the tubes it would be in contact with the coldest cooling water (55°) and hence the condensate would be drawn off at a temperature far below that corresponding to the vacuum.

D. B. Morrison has devised a very ingenious condenser using this principle.[1] His contra-flow condenser, Fig. 100, has a receiver in the base divided off from the rest of the condenser. This receiver is always filled with water which soon reaches the lowest temperature of the cooling water. A two way cock is fitted to the lower part of the condenser so that the condensate can be withdrawn directly from the condenser without passing through this receiver, or it can be made to pass through the

[1] *North-East Coast Inst. of Engineers and Shipbuilders*, 1910.

receiver on its way to the air pump. Thus the temperature of the water and air going to the air pump can be controlled by this valve. In the tropics or elsewhere when there is a fall in vacuum due to an increase in the temperature of the circulating water, he claims that the vacuum can be increased 1½ to 3 in. by use of this device.

With a condenser similar to that described in Art. **169,** the air is cooled in the condenser without lowering the temperature of the condensate. This condenser, therefore, accomplishes practically the same effect of cooling the air as is done by the cold water injection in the dry air cylinder of the Weir air pump.

The wet air pump suction should always lead from the bottom of the condenser and the air pump located well below the condenser so that there will be a pressure head on the pump suction sufficient to lift the foot valves. If an attempt is made to withdraw the air and water vapor at a temperature too close to that corresponding to the vacuum there will be danger of the pump becoming vapor bound.

With high vacuum, the difference between the temperature corresponding to the vacuum and that of the sea water used for cooling water is small. For a 29-in. vacuum this temperature difference is about 20° in the North Atlantic and in southern waters the difference will be smaller and may even disappear. (See Art. **197** for an illustrative example.) A large depression of the air pump suction cannot, therefore, be counted on and the air will often be withdrawn at a fairly large volume and then reduced by cooling or compression in the air pump.

# CHAPTER XVII

## FEED WATER HEATERS

**177. General.**—Feed water heaters are used on practically all merchant and naval vessels today for raising the temperature of the feed water above that of the feed tank. Feed heating not only increases the overall efficiency of the plant but it also increases the evaporation in the boilers. By relieving the boiler of the strains and unequal expansions set up by water at low temperature, feed heating increases the life of the boiler and reduces the maintenance charges. If the feed water was injected into the boiler at the same temperature at which it left the condenser, the large difference in temperature between the water in the boiler and the entering feed would set up severe local strains and disturb the circulation and evaporation in the boiler. For a ship using steam at 250 lbs. pressure and a vacuum of 28½ in., the water in the boiler would be at 406°F. while the condensate entering the feed tank would be in the neighborhood of 80°—a difference of 326°F.

The exhaust steam from the auxiliaries is used almost universally for heating the feed water, although a few old ships are fitted with feed water heaters using live steam. Obviously, there is a loss instead of gain in economy if live steam is used because of the boiler losses in generating the heating steam, although the strains on the boiler due to cold feed water are relieved. By using the heat in the exhaust steam from the auxiliaries there is a direct gain, as heat which would otherwise be thrown away is recovered. Auxiliary machinery on shipboard is always run non-condensing. Non-condensing auxiliaries are cheaper and require less attention than those operating with a vacuum.

For the installation mentioned above operating between 250 lbs. and 28½ in., the saving due to heating the feed water to 220°F. is 140 b.t.u. per pound, or $\frac{140}{1{,}154} = 12.1$ per cent. Not only are 140 b.t.u. per pound or 12.1 per cent of the heat saved, but this

heat does not have to be added in the boiler as heat of the liquid, and the capacity of the boiler is increased.

**178. Open and Closed Heaters.**—There are two types of feed water heaters used in power plant work—open and closed. In open heaters the feed water and the heating steam mix; in the closed or tubular heater the heating steam and feed water do not mix; the water passes through tubes and the heating steam enters the space within the shell and surrounds the tubes on the outside. Figure 101 shows a type of open heater, and Figs. 102 and 103 show types of closed heaters. Both open and closed heaters accomplish the same results in raising the temperature of the feed water. The closed heater acts the same as a surface condenser working without a vacuum. The feed water passing through the tubes condenses the steam and absorbs the heat of vaporization ($x_2r_2$); the heat of the liquid ($q_2$) is carried away by the drip from the feed water heater to the feed water tank and is absorbed by the feed water when the drip mixes with the condensate from the main engines in the feed tank. The open heater acts as a jet condenser. The steam and feed water mix together and the heat of vaporization ($x_2r_2$) and the heat of the liquid ($q_2$) in the entering steam are given up directly to the water when the two mix on entering the heater. The heat of the liquid ($q_2$) mentioned above is the heat above that in the air pump discharge.

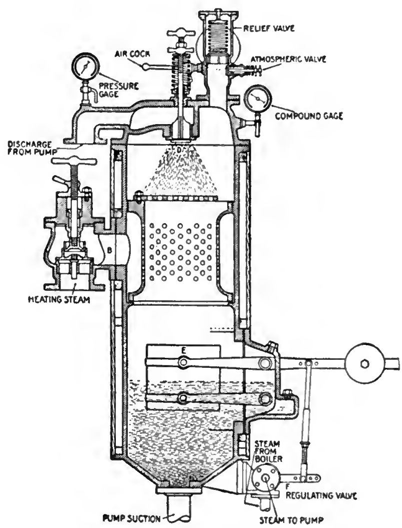



FIG. 101.—Bethlehem-Weir open feed water heater.

Open heaters are cheaper, simpler to maintain, and free the air from the feed water better than closed heaters. The temperature of the feed can be raised slightly higher for a given pressure of heating steam than with closed heaters. On the other hand, closed heaters occupy more space than open heaters and as

they are always located on the suction side of the feed pump, the temperature of the feed water is limited to about 180° unless the heater is given an elevation well above the feed pump. Water at 180°F. is about the hottest that can be handled by a feed pump unless there is a positive head on the suction side of the pump. An additional "hot well" pump is required with

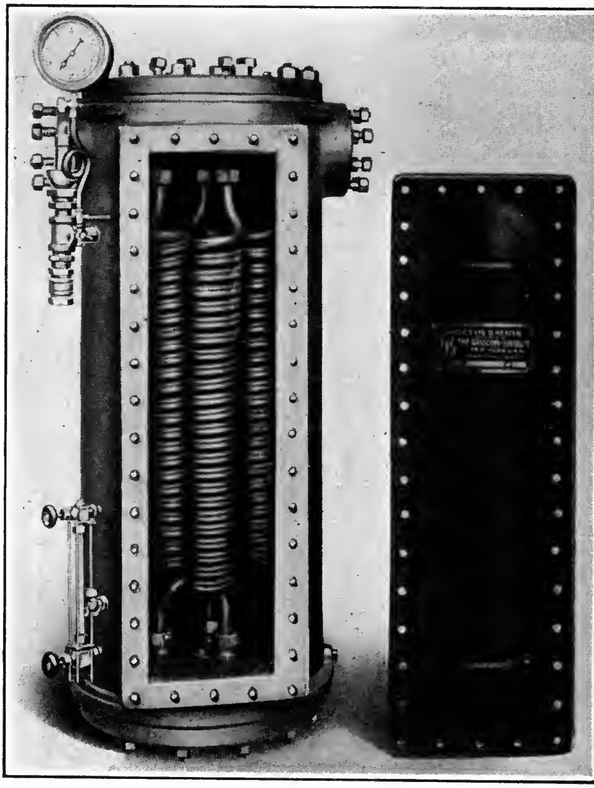


Fig. 102.—Reilly feed water heater.

open heaters to pump the feed water from the feed and filter tank to the heater.

Open heaters are employed in stationary plants more than closed heaters because of their cheapness and because of the more complete separation of the air from the feed water possible with the open heater. They have not been used to any extent on shipboard chiefly because of the large amount of lubricating oil generally found in the auxiliary exhaust. Because of the

more general use of turbine and electric driven auxiliaries and the consequent freedom of oil in the auxiliary exhaust steam, open heaters are being advocated today to some extent for use on shipboard.

Closed heaters may be located either on the suction side (low pressure heaters) or discharge side (high pressure heaters) of the feed pump. The general practice, however, is to fit them on the discharge side of the pump as there is then no difficulty with vapor forming in the pump cylinder due to the high temperature of the pump suction. Heaters located on the discharge side of the feed pump must be built to withstand full boiler pressure and consequently are heavier, more expensive and more difficult to keep tight than low pressure heaters.

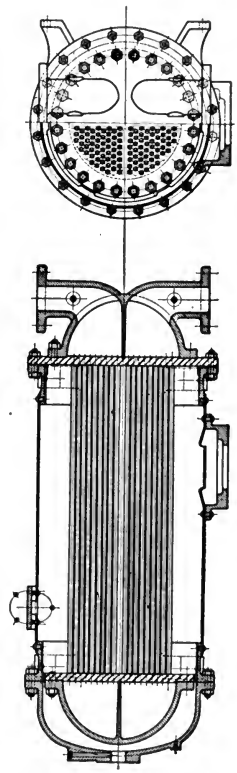
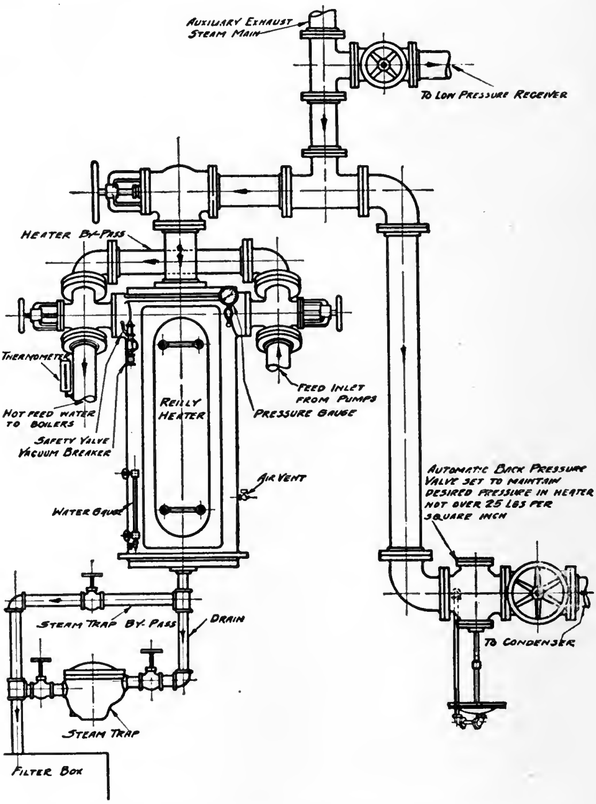


Fig. 103.—Schutte & Koerting feed water heater.

**179. Closed Heater Installation.**—Figure 104a shows an installation of a Reilly two pass closed feed water heater. The auxiliary exhaust line is connected to the shell of the heater, and also through an automatic back pressure valve to the auxiliary condenser. If the supply of exhaust steam is greater than needed for feed heating the back pressure valve will open and discharge steam to the auxiliary condenser as soon as the pressure builds up in the exhaust line. The condensate from the exhaust steam is drained from the shell of the feed heater to the filter end of the feed and filter tank, so that the heat of the liquid above the temperature of the air pump suction is recovered. The drain to the feed and filter tank is either direct or through a trap. In the former case a small amount of water is kept in the bottom of the heater as a water seal. A gage glass is fitted to the bottom of the heater so that the water level can be observed and the correct level is maintained by regulating a valve in the drain line. Air vents are fitted in the bottom of the heater just above the water

surface so that the air that comes in with the exhaust steam can be blown off. This air collects in the lower part of the heater and unless it is periodically blown off the heater will fill up with air and fail to function. An accumulation of air in the heater will cut down the tube area, resulting in a lowering of the temperature of the feed water leaving the heater.




BY PERMISSION GRISCOM-RUSSELL CO.

FIG. 104a.—Diagram of two-pass induction type installation.

Heaters are often connected up as shown in Fig. 104b, with the automatic back pressure valve on the discharge side of the heater. All the steam in the exhaust line passes through the heater but if there is an excess of heating steam part of it will be discharged to the auxiliary condenser instead of being condensed in the heater.

The thoroughfare type has the advantage in that there is no

tendency for the feed heater to become air bound as might be the case with the induction type if the air vent was not given proper attention.

**180. Types of Closed Feed Heaters.**—Figure 102 shows a Reilly marine type two pass heater. This make of heater is built in both single and double pass. The single pass is lighter and somewhat simpler; the double pass, however, has a higher

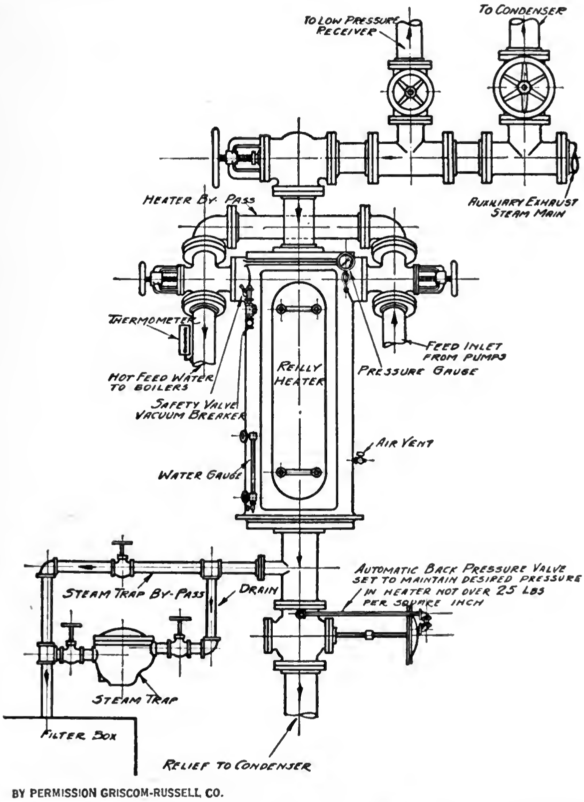



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Fig. 104b.—Diagram of two-pass thoroughfare type installation.

efficiency as the length of the water travel in the steam space is twice as great as in the single pass type. The heating surface of the Reilly heater consists of helical coils fastened to the top and bottom headers by screwed union joints. The helical shape of the tubes causes the water in its passage through them to be constantly thrown to the outside by centrifugal force. The tubing is also crimped, which further assists to agitate the water

in its passage along the tube. Because of the high efficiency of the tubes in heating the water, a much smaller heating surface is required than when straight tubes are used. The Reilly heater has no baffles in the steam space, which are required in all straight tube heaters. This heater is built with a cast iron shell which insures a long life to the heater.

Figure 103 shows a Schutte & Koerting four pass feed water heater. The feed water enters at the top, makes four passes through the tubes as shown in Fig. 103 and leaves at the top. The shell is of cast iron and the tubes are $\frac{5}{8}$ in. seamless drawn of brass or copper. The exhaust steam enters the shell near the top and drains off at the bottom. The construction, as clearly shown in Fig. 103, allows the heater to be readily taken apart for inspection or replacing a tube.

**181. Series Heaters.**—The temperature of the feed water leaving the feed heater can never be greater than the temperature corresponding to the pressure of the exhaust steam and generally the feed temperature is 5 to 12° lower than the auxiliary exhaust. With auxiliary exhaust at 5 lbs. gage (228°F.) the feed temperature is around 220°F. It often happens that there is more auxiliary exhaust steam than can be used for feed heating (see example in Art. **205**) and under such conditions this heat is frequently rejected to the auxiliary condenser as a dead loss.

In order to increase the feed temperature, series heaters are sometimes used. Secondary heaters are placed between the main heater and the boiler or series coils are placed in the discharge end of the main feed heater. These receive the discharge from the direct acting pumps or the drips from the oil heater, evaporator, etc., which are at a higher temperature than the auxiliary exhaust. While this will increase the temperature of the feed water and the efficiency of the plant the increased complications caused by the additional piping and separate exhaust lines seldom warrant the adoption of series heaters.

**182. Economizers.**—An economizer is a feed water heater located in the path of the hot gases between the boiler and the stack. They are widely used in land installation and show a good gain in economy by transferring the heat of the waste gases to the feed water. When used on shore mechanical draft is always required as economizers not only create an added

resistance but reduce the stack temperature to such a low figure that natural draft is impossible.

The Howden draft system as used with Scotch boilers is similar in principle; but here the heat in the waste gases is used to heat the air used for combustion instead of the feed water.

In land installations where economizers are used a large supply of auxiliary exhaust steam is not available for feed heating as on shipboard. As shown in the calculations for feed heater (Art. **205**), there is generally sufficient exhaust steam on shipboard to raise the feed water up to a temperature of 220°F. Therefore, if economizers are used on shipboard they must raise the feed water from around 220°F. to as high a temperature as possible with the waste gases.

The feed water is pumped through the economizer under full boiler pressure; the water enters the economizer at the upper part so that the coldest water is in contact with the coolest gases and the flow of the feed water is thus opposite to that of the gases.

An example will show the possibilities of the economizer:

Boiler heating surface = 5,240 sq. ft.
Actual evaporation per hour = 33,000 lbs.
Oil per hour = 2,630 lbs.
Temperature of feed entering economizer $t_1 = 220°F.$
Assumed temperature of feed leaving economizer $t_2 = 300°F.$
Temperature of gases leaving boiler $t_3 = 550°F.$
Assumed temperature of gases leaving economizer $t_4 = 320°F.$
Mean temperature difference between gases and water $t_m$.

$$t_m = \frac{250 - 100}{\log_e 2.5} = 164°\text{F}.$$

Stack gases with 24 per cent excess air $= (14 \times 1.24) + 1.0 = 18.3$ lbs. per lb. oil.

Specific heat of gases = .238.

B.t.u. available in waste gases $= 18.3 \times 2630 \times .238 \times 230° = 2{,}640{,}000$ per hour.

B.t.u. required to raise feed water (80°) $= 33{,}000 \times 80 = 2{,}640{,}000$ per hour.

The cônditions have been assumed so as to give a balance between heat available in the gases and heat required to raise the feed water. Thus under the assumed conditions we can expect a rise of 80° in the feed water for a drop of 230° in the temperature of the waste gases.

The coefficient of heat transfer between gases and water is about 4.5 b.t.u. per sq. ft. per degree difference per hour. B.t.u. heat transfer per sq. ft. per hour $=4.5\times164=740$ b.t.u.

$$\text{Economizer heating surface required}=\frac{2,640,000}{740}=3,600 \text{ sq. ft.}$$

As the heating surface of the boiler is 5,240 sq. ft., we see that the economizer surface necessary to increase the feed temperature 80° is nearly 70 per cent of that required for generating steam in the boilers. It is evident, as shown by the above example, that both space and weight requirements prohibit the use of economizers on shipboard unless used in the place of the auxiliary exhaust method of heating the feed water.

If auxiliaries with low steam consumption are used, as would be the case with electric driven auxiliaries supplied with current from a turbine driven generator, economizers are entirely feasible. With such an installation the steam consumption of the auxiliaries would be about one-half what it is at present and the temperature of the feed water leaving the heater would be around 140°F. Under these conditions the economizer would operate with a greater mean temperature difference and its heating surface would be smaller.

There is no doubt that attention to economy will next be focused on the auxiliaries. It is certainly poor engineering to heat the feed water with steam when a large supply of heat is available in the uptake gases. A single generating set working under high pressure and at the same vacuum as the main turbine and supplying current to electrically driven auxiliaries is an obvious step in marine engineering.

# CHAPTER XVIII

## THE POWER PLANT LAYOUT

**183. Power Plant Diagram.**—A diagrammatic layout of the marine power plant is shown in Fig. 105. The live steam line, exhaust steam lines and feed lines are clearly shown by different types of lines. In order not to make the diagram too complicated several smaller pieces of apparatus, as the forced draft blower and auxiliary circulating pump, have been omitted and the evaporator and distiller auxiliaries have also been left out. The diagram should receive careful attention during the study of marine engineering and especially in connection with the power plant calculations in Chap. XIX.

*Power Plant Layouts.*—Typical engine and boiler room layouts are shown in Figs. 106 to 112. Figures 106a and 106b show a plan and elevation of the machinery of the electric driven cargo ship "Eclipse," one of the first merchant ships to be fitted with this type of machinery. Figure 107 shows the electric drive layout for the quadruple screw battleship "New Mexico." The machinery arrangement for the battle cruiser "Hood" is shown in Fig. 111. This installation consists of four single reduction gear units of 36,000 s.h.p., the data for which are given in Table X. Figure 108 shows the echelon arrangement of the geared turbine machinery of the destroyer "Wadsworth" and Fig. 112 the arrangement of a ship with geared electric drive (Table X). Figures 109 and 110 are included as a representative layout of a single screw geared turbine ship.

Layouts of merchant ship machinery with reciprocating engines, geared turbines, etc., have not been included as the reader can find such layouts in current issues of marine engineering publications.

The above layouts need no explanation. They are included here to show some of the special types of machinery used for ship propulsion.

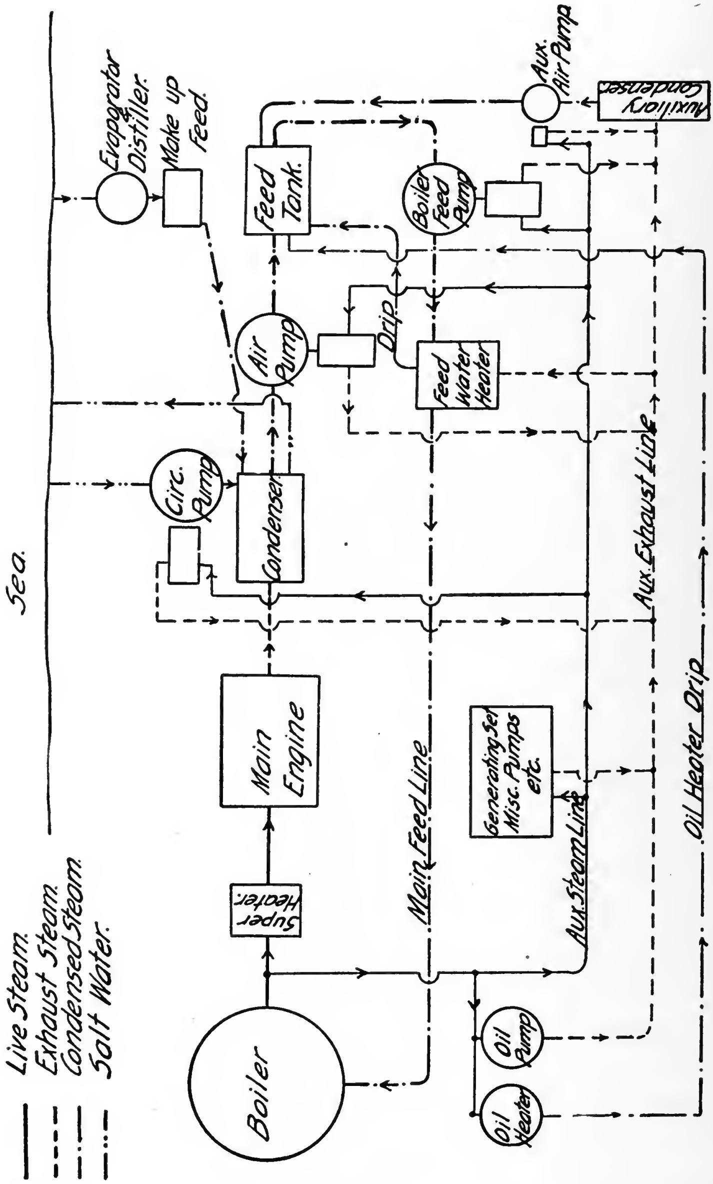
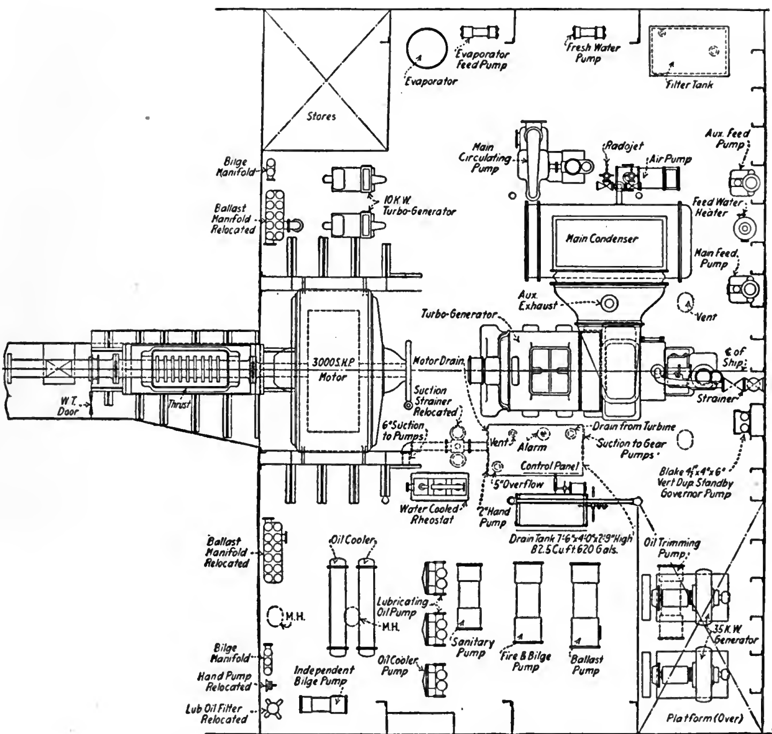



FIG. 105.—Power plant diagram.




BY PERMISSION "MARINE ENGINEERING"

Fig. 106a.—Plan of engine room of the "Eclipse."

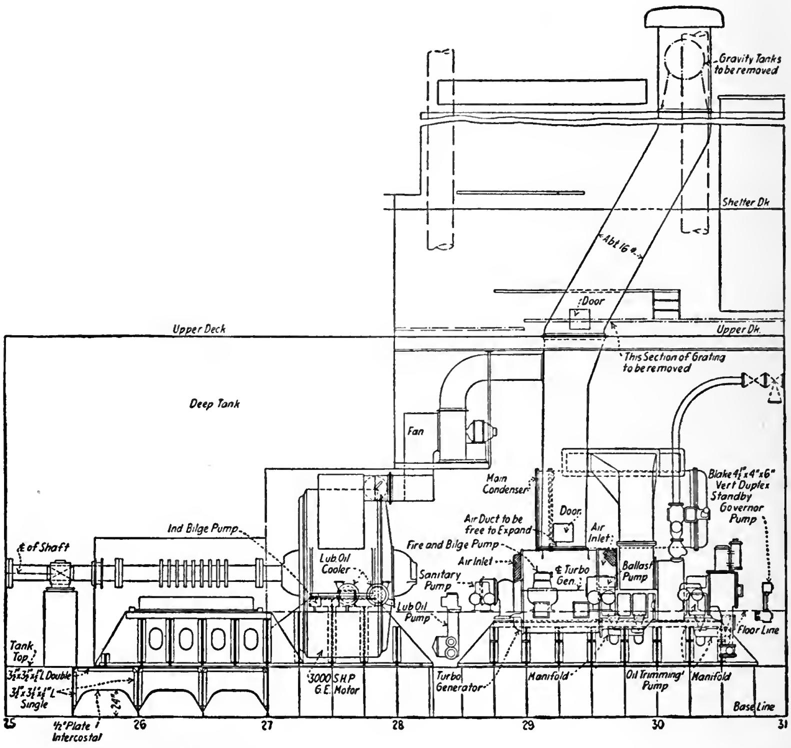



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FIG. 106b.—Elevation of engine room of the "Eclipse."

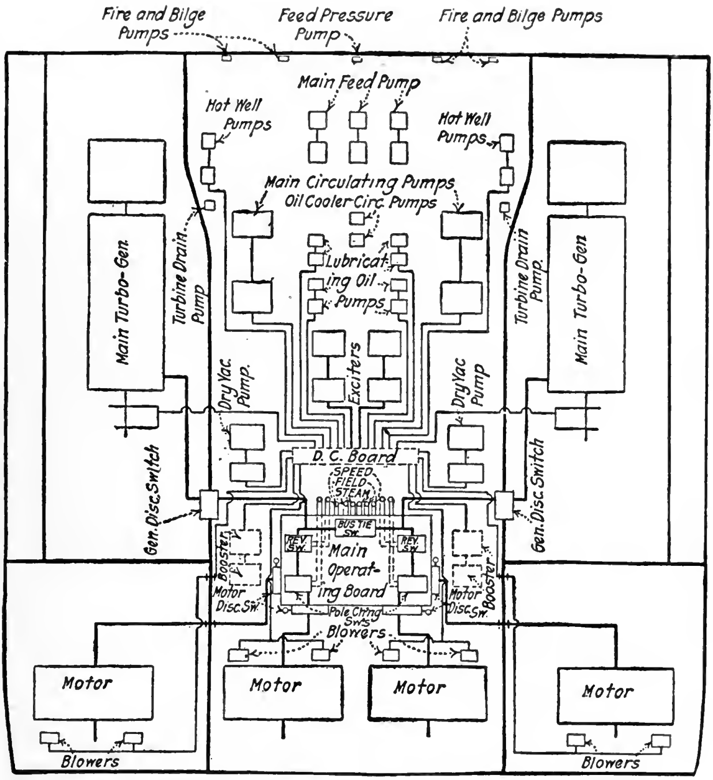



Fig. 107.—Machinery arrangement of U. S. S. "New Mexico."

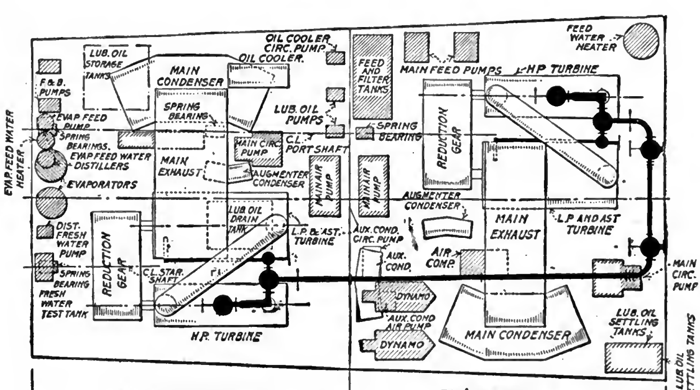



Fig. 108.—Machinery arrangement U. S. S. "Wadsworth."

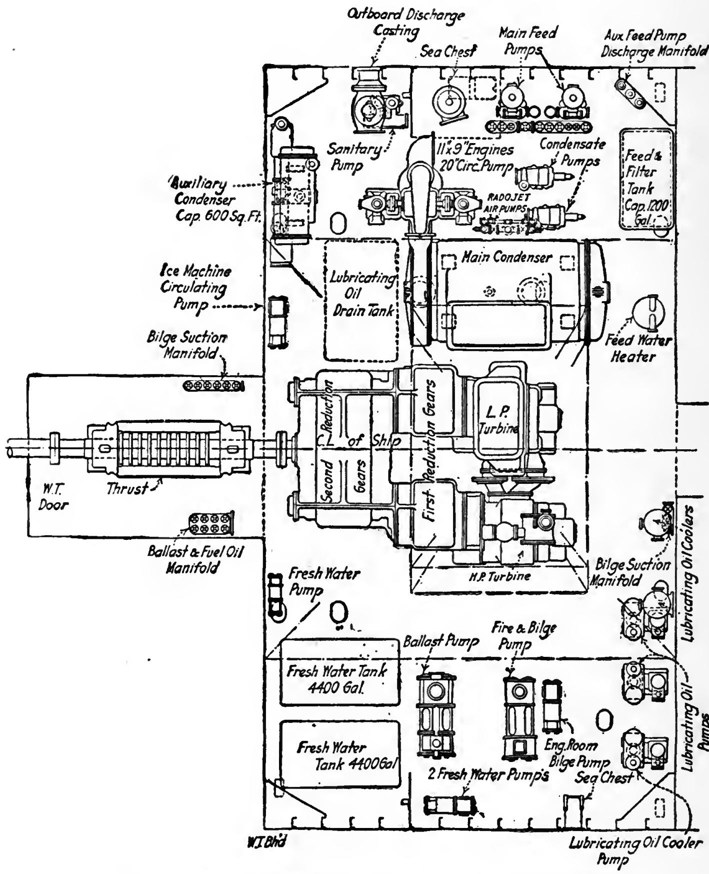



FIG. 109.—Plan of engine room of geared turbine ship.

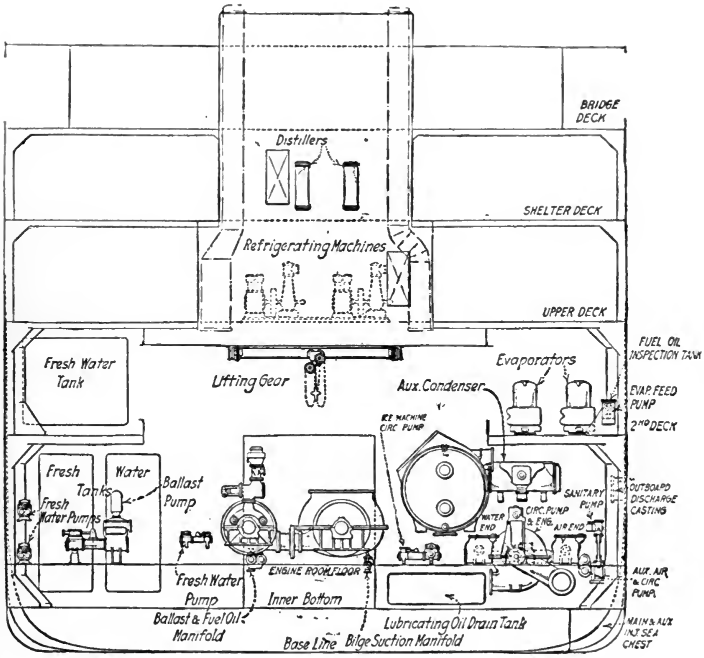
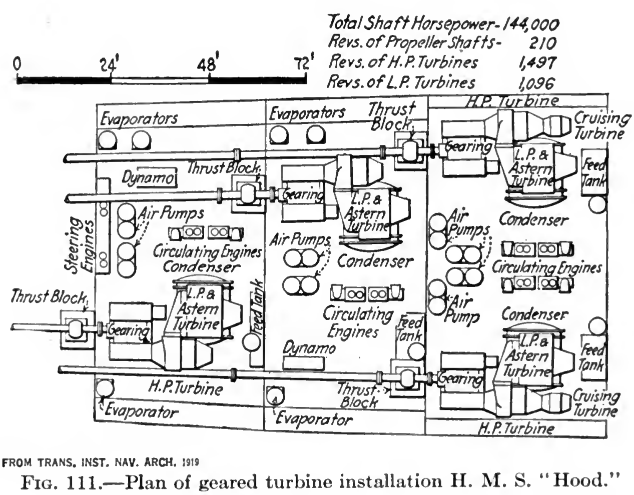



FIG. 110.—Section through engine room shown in figure 109.




FROM TRANS. INST. NAV. ARCH. 1919

FIG. 111.—Plan of geared turbine installation H. M. S. "Hood."

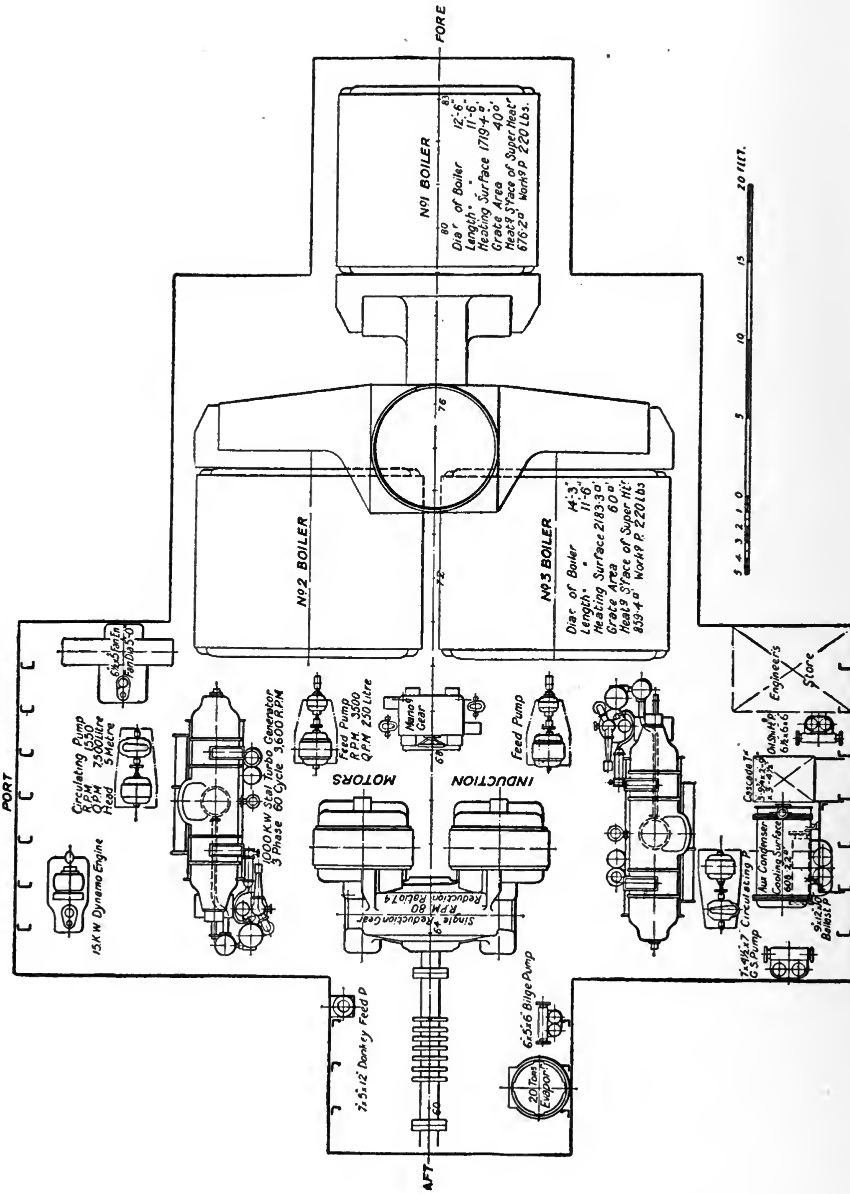



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FIG. 112.—Machinery arrangement of S. S. "Biyo Maru" geared electric drive.

## CHAPTER XIX

## COMPUTATIONS FOR THE POWER PLANT OF THE MERCHANT SHIP

*Case I.—Geared turbines with both coal and oil fired water-tube boilers.*

E.H.P. = 1,500.
Service s.h.p. = 2,700 (single screw).
Service speed = 11 knots.
Propeller r.p.m. = 90.
(See *Journal A. S. N. E.*, May, 1920, for propeller calculations.)
Double reduction gears with reduction of 36 : 1.
Steam pressure at throttle = 260 lbs. gage.
Boiler pressure = 270 lbs. gage.
Superheat = 100°F.
Vacuum in condenser 29 in. (30 in. barometer).
Back pressure at turbine exhaust nozzle = 28.8 in.

High initial steam pressure with a reasonably high superheat and a high vacuum has been adopted in order to reduce the steam consumption as low as possible. A high vacuum requires close attention to air leakage, but with a reasonably tight system, proper air pump equipment and an efficient engine force, 29 in. would be possible except in tropical waters. There is no doubt that 29 in. is rather too high to expect in service conditions on the average cargo steamer. It has been selected more to illustrate the possibilities and value of high vacuum than to follow present day practice.

A steam pressure as high as 270 lbs. is beyond the practice of Scotch boilers, but good practice with water-tube boilers. In Case II, where Scotch boilers are used, a lower initial pressure will be used.

**184. Steam Consumption of Main Turbine.**—The main engines should be considered first so that a steam consumption can be settled on for estimating the size of boilers and auxiliaries. Needless to say, a fairly accurate estimate of the steam con-

sumption should be made, for the heating surface of the boilers will depend directly upon this figure. It is assumed that, at this stage in the design, no guarantee is available from the manufacturer.

The proper procedure is to calculate the steam consumption of the theoretical Rankine cycle between the limits adopted and then to correct this by the ratio of the theoretical steam consumption to actual steam consumption of a similar installation (Art. **17**).

The steam consumption of the Rankine cycle is

$$W_t = \frac{2,545}{H_1 - H_2}$$

where $H_1$ = total initial heat contents of steam $= q_1 + r_1 + ct_s$.

$H_2$ = heat contents at exhaust pressure after adiabatic expansion.

$= x_2 r_2 + q_2$.

For the turbine under consideration:

$$\begin{aligned} H_1 &= 1,264 \text{ at 275 lbs. abs. and } 100^\circ \text{ superheat.} \\ H_2 &= \ \ 852 \text{ at 29-in. vacuum (.49 lb./sq. in.)} \\ \hline H_1 - H_2 &= \ \ 412 \end{aligned}$$

$$W_t = \frac{2,545}{412} = 6.2 \text{ lbs. steam per s.h.p. per hour.}$$

(Data from Peabody's Mollier Diagram)

If the efficiency ratio from the trial of a similar turbine is available we can apply it to the above theoretical water rate and have a close estimate of the steam consumption of the turbine.

Efficiency ratios from actual trials and tests vary between .50 and .68.[1] If values of efficiency ratio are not available they

[1] Efficiency ratios for stationary installation are as a rule higher than for marine installations. As pointed out in Art. **104**, values as high as .73 have been recorded on shore.

The efficiency ratio for a geared turbine installation includes the loss in the gears as the S.H.P. is always measured beyond the gears. Efficiency ratios for turbines driving generators (electric drive) will always run higher than for geared turbines for the above reason. (See table of comparison in Art. **162**.)

may be worked up from trial data as shown in the following examples:

Published data of a trial of a similar turbine of 4,000 s.h.p. showed a steam consumption on trial of 11.2 lbs. per s.h.p. per hour with an initial steam pressure of $P_1 = 191$ lbs. abs. dry steam and a vacuum of 28.75 lbs. ($P_2 = .61$ lbs./sq.in.)

The calculations for the theoretical steam consumption of this turbine are as follows:

$$\begin{aligned} H_1 &= 1{,}197 \text{ at 191 lbs. abs.} \quad \text{No superheat} \\ H_2 &= \underline{\ \ 840} \\ H_1 - H_2 &= \ \ 357 \end{aligned}$$

$$W_t = \frac{2{,}545}{357} = 7.14 \text{ lbs.}$$

Ratio of theoretical to actual (efficiency ratio) $= \dfrac{7.14}{11.2} = .64.$

As a further illustration of the use of efficiency ratio the following comparison is given. In *Marine Engineering*, for May, 1919, a proposal for an electric drive[1] was given. The estimated steam consumption of the turbine was quoted as 9.4 lbs. with $P_1 =$ 280 lbs. abs. and 150° superheat and $P_2 = 28.5$ in.

$$\begin{aligned} H_1 &= 1{,}292 \text{ at 280 lbs. abs.} \quad 150° \text{ superheat.} \\ H_2 &= \underline{\ \ 887} \text{ at .75 lbs./sq.in. abs.} \\ H_1 - H_2 &= \ \ 405 \end{aligned}$$

$$\text{Eff. Ratio} = \frac{2{,}545}{W_a\,(H_1 - H_2)} = \frac{2{,}545}{9.4 \times 405} = .67$$

By estimating the steam consumption by the procedure outlined above all guesswork is eliminated and a proper allowance can be made for changes of superheat, increase or decrease of initial steam pressure, and variations in vacuum. The theoretical result of course is of little value unless coefficients are available from practice.

Returning again to the problem in hand and using an efficiency ratio of .62 we have:

Actual steam consumption ($W_a$) $= \dfrac{6.2}{.62}$ lbs. $=$ 10.0 lbs. s.h.p./hour.

[1] See footnote on page 280.

This of course is an estimated figure and may have to be modified later to agree with the manufacturer's guarantee. Obviously for continuous service at sea, where the conditions are not those that exist on trials and where air leakage may reduce the vacuum, a steam consumption as low as obtained on trial cannot be expected. The efficiency of .62 used above is very conservative for trial conditions and later a percentage allowance will be used to cover all possible increases in steam and fuel consumption in service.

**185. Total Steam Consumption.**—In order to determine the size of the boilers and other auxiliaries an estimate must be made at this stage of the steam used by the auxiliaries.

The present practice is to quote auxiliary steam consumption as a percentage of the main engines. This is not a good method, especially for an installation like the one under consideration. The steam consumption of many of the auxiliaries is a constant quantity and but few show a reduction in steam consumption with a reduction in main engine steam; in fact some, such as the circulating and air pump, will increase with a high vacuum of 29 in. Consequently, with a geared turbine of low steam consumption and high vacuum we would expect a higher percentage than with a reciprocating engine. A much better way to quote auxiliary consumption is in pounds per s.h.p. of the main engines.

As will be shown later no effort is going to be made in this installation to use auxiliaries of high efficiency, and a rather high figure should be used for auxiliary steam. A reasonable value for this, from data at hand, is about 20 per cent of that used by the main engines or 2.0 per s.h.p. of main engines per hour.

We now have for the estimated total steam consumption:

| | |
|---|---|
| Main engines: | $2{,}700 \times 10 = 27{,}000$ lbs. |
| Auxiliaries: | $27{,}000 \times .20 = 5{,}400$ lbs. |
| Total steam. . . . . . . . . . . . . | $= 32{,}400$ lbs. per hour. |

Later a separate calculation will be made for the steam used by each piece of auxiliary machinery and a more accurate value obtained for auxiliary steam consumption.

With the above estimate of steam consumption we are now

ready to calculate the size of boilers required. Water-tube boilers will be used in this case and, to fully illustrate the problem, calculations will be carried through for both coal fired and oil fired boilers.

| | Lbs. |
|---|---|
| Steam used for propelling machinery and ship auxiliaries | 32,400 |
| Arbitrary allowance for heating | 600 |
| Total steam to be generated per hour | 33,000 |

$$\text{Steam per s.h.p. per hour all purposes} = \frac{33,000}{2,700} = 12.2 \text{ lbs.}$$

**186. Boilers.**—(*a*) *Babcock & Wilcox coal fired water-tube boilers.*—Experience with hand fired boilers in the merchant service indicates 20 to 25 lbs. of coal per sq. ft. of grate is about the maximum that can be attained for continuous service without undue difficulty in stoking. Probably 22 lbs. of "dry coal" per sq.ft. g.s. is as high a figure as should be used for design purposes. Higher values of this can be maintained for short times; 30 and even 45 lbs. per sq.ft. G.S. per hour have been used on naval vessels but not for continuous service, however.

It will be observed that the above rate is for "dry coal" with a reasonable percentage of ash. If the coal contains 5 per cent moisture the rate will be 23 lbs. per sq. ft.; and if the coal is high in ash an even higher rate will have to be maintained to give the required evaporation.

It is not a wise plan to cut too close on boiler heating surface or allow too high a rate of combustion, for a ship that requires hard firing to maintain the required steam pressure soon gets a bad reputation and it is more difficult to pick up firemen.

The factor of evaporation for a boiler producing steam at 285 lbs. abs. and 100° superheat and feed at 218°F. (10° below auxiliary back pressure of 5 lbs. gage) is as follows:

$$\text{Factor of Evaporation} = \frac{1,265 - 186}{970.4} = 1.11$$

The size of the boiler is obtained by the following procedure:

Assume heat contents of coal = 14,000 b.t.u. per lb.
Assume a boiler efficiency in service = 66 per cent.

Evaporation per hour (actual) = 33,000 lbs.

Assumed rate of combustion = 22 lbs. dry coal per hour per sq.ft. G.S.

Coal required per hour

$$= \frac{33{,}000 \times 1{,}079}{14{,}000 \times .66} = 3{,}860 \text{ lbs.}$$

where 1,079 = (1,264 − 186) the heat absorbed by one lb. of steam $(H_1 - q_2)$, (see factor of evaporation calculation).

Grate surface required with a rate of combustion of 22 lbs.

$$= \frac{3{,}860}{22} = 175 \text{ sq. ft.}$$

Heating surface required with ratio $\frac{\text{H.S.}}{\text{G.S.}}$ of 37.5 = 6,550 sq. ft.

Actual evaporation per sq. ft. heating surface per hour

$$= \frac{33{,}000}{6{,}550} = 5.03 \text{ lbs.}$$

Equivalent evaporation per sq.ft. of H.S. per hour = 5.03 × 1.11 5.58.

The foregoing method of computing the factor of evaporation and the coal required per hour is not absolutely correct. The assumption has been made that all the steam is superheated to 100°F. The auxiliary steam and the heating steam are not superheated but are withdrawn between the boiler and the superheater as saturated steam at 270 lbs. gage. This steam is then reduced to 100 lbs. gage before entering the auxiliary steam line. During the reducing, which is a constant heat change, the steam will be slightly superheated. We can neglect this as it will be lost by radiation before reaching the auxiliaries. Hence the actual coal required per hour

$$= \frac{6{,}000 \times (1{,}203 - 186) + 27{,}000 \times (1{,}203 + 62 - 186)}{14{,}000 \times .66} = 3{,}800 \text{ lbs.}$$

It will be observed that the error is 60 lbs. in 3,800 lbs. or about 1.5 per cent. The arbitrary method of selecting boiler effi-

ciency and heat contents of the coal would not warrant the above refinement in calculations at this stage. However, the actual condition should be kept in mind, for in certain cases it might be of importance.

The rate of evaporation calculated above can be compared with the curve shown in Fig. 113, as a check. The curve for the B. & W. boiler in Fig. 113 shows that for a rate of combustion of 22 lbs. of dry coal per sq. ft. G.S. per hour the equivalent evaporation per sq. ft. of heating surface is 5.6 lbs. In this particular case the ratio of H.S./G.S. was chosen so that it gave an exact check with the curve. This, however, will not always happen.

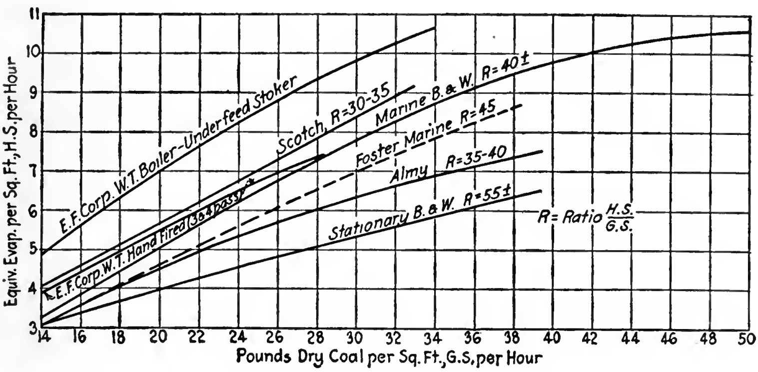


Fig. 113.

The curves in Fig. 113 have been compiled from a large number of tests and will serve as a guide for fixing the heating surface of coal fired boilers. They should be used with care and discretion, however, for the evaporation per sq. ft. of H.S. will vary not only with changes in the ratio of H.S./G.S. but also with the efficiency of the boiler.[1] The fact that the Scotch boiler curve lies above some of the other curves is simply due to the general practice of building Scotch boilers with a lower ratio of H.S./G.S. and does not indicate a higher efficiency.

In order to allow a good margin for an increase in steam

[1] The actual evaporation per sq. ft. of H.S. per hour can be expressed as follows:

$$\text{Actual Evap.} = \frac{\text{Eff.} \times \text{rate of combustion} \times \text{b.t.u. per lb. coal}}{(\text{H.S.} \div \text{G.S.}) \times (H_1 - q_2)}$$

consumption of main engines and auxiliaries in service over that of trial conditions, and to allow for poor coal, bad firing and a general falling off in boiler efficiency this calculated heating surface should be increased by about 15 per cent. This is the only increase so far allowed, for the assumed steam consumption was based on a trial result and the heating surface was based on good service practice. It is far better to hold closely to the calculated figures and allow an overall increase in one place to take care of all contingencies, than to keep adding small increases to each calculation.

Heating surface to be provided $= 6{,}550 \times 1.15 = 7{,}540$ sq. ft. For the boilers with 15 per cent increase in heating surface:

$$\text{H.S.} = 7{,}540 \text{ sq.ft.}$$

$$\text{G.S.} = 201 \text{ sq.ft.}$$

Evaporation per pound of coal (actual)

$$= \frac{33{,}000}{3{,}860} = 8.55 \text{ lbs.}$$

Evaporation per pound of coal (equivalent)

$$= 8.55 \times 1.11 = 9.5 \text{ lbs.}$$

As a check on the calculated equivalent evaporation we have:

$$\frac{14{,}000 \times .66}{970.4} = 9.5 \text{ lbs. of water from and at } 212° \text{ per lb. coal.}$$

From Sterling's "Marine Engineers' Handbook," page 733, the following B. & W. boilers have been tentatively selected. A final selection can only be made by making a study of the arrangement and layout on the plans of the ship.

Three Boilers (4-in. tubes)

Grate surface $= 66.8$ sq.ft. each (3 boilers $= 200$ sq.ft.)
Heating surface $= 2{,}490$ sq. ft. each (3 boilers $= 7{,}460$ sq. ft.)
Ratio $\frac{\text{H.S.}}{\text{G.S.}} = 37.28$.
Length of tube 10 ft.
Weight, dry $= 78{,}000$ lbs. each (excluding superheater).
Weight, wet $= 93{,}000$ lbs. each (excluding superheater).

Weight, 3 boilers = 280,000 lbs. (wet) (excluding superheater).
Weight per sq.ft. H.S. = 37.3 lbs. (excluding superheater).

The value of 66 per cent assumed for efficiency is a fair value for a coal fired water-tube boiler for good service conditions. With efficient firing and boilers equipped with soot blower a boiler efficiency as high as this could be relied on.

$$\text{Coal per s.h.p. per hour} = \frac{3{,}860}{2{,}700} = 1.43 \text{ lbs.}$$

Coal per grate surface per hour with 12.2 lbs. steam per s.h.p. per hour $= \dfrac{3{,}860}{200} = 19.3$ lbs. (G.S. and H.S. increased 15 per cent).

**189. Boiler Draft.**—The amount of draft between furnace and ash pit necessary to burn 22 lbs. dry coal per sq. ft. of grate surface per hour is .30 lb. (from Fig. 38). To this must be added a further loss of .20 in. in the boiler passages, which results in a total required draft of .50 in. Sterling's Handbook, page 346, gives a curve based on the performance of B. & W. boilers and this shows a draft of 1/2 in. required which checks the above estimate.

If a rate of combustion as high as 22 lbs. is to be maintained in service, forced draft becomes almost a necessity as the minimum figure of 1/2 in. draft is rather high for natural draft. If we assume the temperature of the gases leaving the boiler to be 520° (110° higher than the temperature of the saturated steam at 275 lbs. abs.), and the temperature of the outside air to be 80°F.,

$$\text{Height of stack} = 1.25\ P \div \left(\frac{7.63}{540} - \frac{8.05}{980}\right) = 106 \text{ ft.}$$

(Art. 62)

(See approximate rule, Sterling, p. 345.)

A stack height of 100 ft. is rather excessive for a ship of this size; hence forced draft will have to be used or the rate of combustion lowered and larger boilers used. If the stack temperature was increased to 700 or 800°F. the required draft could be obtained with a stack of reasonable height. In such a case, however, the draft would be obtained at the expense of fuel consumption, which would not be good engineering.

Sample calculations for the size and horsepower of a forced draft fan are given under Case (*b*) following:

(*b*) *Oil fired water-tube boiler* (Foster type). A boiler using oil fuel can be operated at higher rate of combustion than with coal as there is no arduous work on the part of the firemen to account for. The rate of combustion is expressed in lbs. of oil burned per sq. ft. heating surface per hour. For a cargo ship a rate of .50 to .60 lb. oil/sq. ft. H.S. is probably high enough to use for continuous service. Rates as high as 1.0 lb./sq. ft. H.S. have been used on naval vessels but not for continuous service, however.

Allowing an efficiency in service of 72 per cent a rate of combustion of .50 lb. oil per sq. ft. H.S., and 18,800 b.t.u. per lb. of oil, we have,

$$\text{Oil burned per hour} = \frac{33{,}000 \times 1{,}079}{18{,}800 \times .72} = 2{,}620 \text{ lbs.}$$

A rate of combustion of 0.50 lb. of oil per sq. ft. of heating surface per hour will give for the required heating surface,

$$\frac{2{,}620}{.50} = 5{,}240 \text{ sq. ft. H.S.}$$

$$\text{Actual evaporation per lb. of oil} = \frac{33{,}000}{2{,}620} = 12.6 \text{ lbs.}$$

$$\text{Equivalent evaporation per lb. of oil} = 12.6 \times 1.11 = 14.0 \text{ lbs.}$$

$$\text{Equivalent evaporation per sq. ft. H.S.} = \frac{33{,}000}{5{,}240} \times 1.11 = 7.0.$$

$$\text{Oil per s.h.p. per hour} = \frac{2{,}620}{2{,}700} = 0.97 \text{ lb.}$$

Heating surface of boilers to be installed $= 5{,}240 \times 1.15 = 6{,}020$ sq. ft. Approximate draft required $= 1.00$ in. water. (From tests on this type of boiler.)

The amount of draft to use with fuel oil is a rather illusive factor, for published data vary widely. Ships of the Navy with closed stokeholes use a much higher rate, for military reasons, than is used in the merchant service. The fan should be capable of delivering the proper amount of air at 2 in. of water—twice the pressure given above.

**188. Draft Apparatus.**—The theoretical amount of air per

pound of fuel oil is about 14 lbs. (184 cu. ft.) for the average run of fuel oil. Allowing 30 per cent excess air, the air required per hour is:

Air per hour $= 2{,}620 \times 14 \times 1.30 = 47{,}500$ lbs.

Blower capacity in service =

$$\frac{47{,}500}{60 \times .076} = 10{,}400 \text{ cu. ft. per min. at } 60°\text{F.}$$

Air pressure allowed = 1.0 in. of water (68 ft. air).
Service hp. of blower at 40 per cent efficiency =

$$\frac{47{,}500 \times 68}{33{,}000 \times 60 \times .40} = 4.0 \text{ hp.}$$

The common practice in fixing the area of stack and uptakes is to allow 1 sq. ft. of area for each 200 lbs. oil burned per hour. Naval vessels use smaller stacks than this, but on the other hand the drafts used are greater.

$$\text{Area of stack} = \frac{2{,}620}{200} = 13.1 \text{ sq. ft.}$$

*Case II. Scotch Boilers with Coal; Reciprocating Engines.* The following conditions are assumed for this case:

I.H.P. = 2,900
R.p.m. = 90
$P_1$ at throttle = 200 lbs. gage (215 lbs. abs.)
Boiler pressure = 210 lbs. gage (225 lbs. abs.)
Superheat at boiler = 60°F. ($t_1 = 452$°F.)
Vacuum in condenser = 27 in.
Four cylinder triple expansion engine.

**189. Main Engines (steam consumption).**—We will follow the same procedure to find the steam consumption as was used in Case I. From the trial data of the U.S.S. "Delaware" the efficiency ratio is found to be 56 per cent. Calculations both for the "Delaware" and the assumed conditions follow:

*U.S.S. "Delaware:"*

$P_1 = 285$ lbs. abs. (at engine)
Superheat = 61.6°F. (at engine)

Vacuum = 26.3 in. (1.82 lbs. abs.)
Steam consumption on trial ($w_a$) = 13.38 lbs. per i.h.p. per hour.

$$(w_t) \text{ theoretical consumption } = \frac{2545}{340} = 7.50 \text{ lbs. per i.h.p.}$$

$$H_1 = 1{,}245$$
$$H_2 = \underline{\phantom{0}905}$$
$$H_1 - H_2 = \phantom{0}340$$

$$\text{Eff. ratio} = \frac{7.50}{13.38} = .56$$

The above data are from the full power trial; at 19 knots the efficiency ratio was higher (see A. S. N. E., Nov., 1909). For further data on efficiency ratios see Table IX.

Steam consumption for Case II:

$P_1$ at throttle = 215 lbs. abs.
Superheat (at throttle) = 55°
Vacuum in condenser = 27 in. (1.47 lbs.)
Assumed efficiency ratio = 55 per cent

$$H_1 = 1{,}232$$
$$H_2 = \underline{\phantom{0}902}$$
$$H_1 - H_2 = \phantom{0}330$$

$$\text{Theoretical steam consumption} = \frac{2545}{330} = 7.70 \text{ lbs. per i.h.p. per hour.}$$

$$\text{Probable action consumption} = \frac{7.70}{.55} = 14.0 \text{ lbs. per i.h.p. per hour.}$$

Before taking up the consideration of the boilers it is necessary to make an estimate of the steam used by auxiliaries. Naval vessels with engines similar to the "Delaware" show a steam consumption for the main engine auxiliaries of about 12 per cent of that of main engines. This, however, does not include generator set or hull auxiliaries. Merchant ship auxiliaries use about 15 per cent of that used by the main engines or about 2.0 per i.h.p. per hour. Allowing 15 per cent for the installation under consideration we have 6,100 lbs. per hour or 2.1 lbs. per i.h.p. per hour. As pointed out in Case I a detailed estimate should be made for each auxiliary later on, to check up this preliminary estimate.

| | Lbs per hour |
|---|---|
| Steam required for main engine = 2,900 × 14 = | 40,600 |
| Steam required for all auxiliaries (15 per cent) | 6,100 |
| | 46,700 |
| Heating (arbitrary allowance) | 600 |
| Total steam required per hour (16.3 lbs. per I.H.P.) | 47,300 |

**190. Scotch Boilers.**—We will assume the same rate of combustion as was used in Case I with water-tube boilers, namely, 22 lbs. dry coal per sq. ft. of grate surface per hour. This is close to the upper limit for continuous hand firing. We will, however, design the boiler for this rate and then, as before, allow sufficient margin in the heating surface so that this rate will not be exceeded in service.

The factor of evaporation for the conditions stated is

$$\text{Factor of evaporation} = \frac{H_1 - q_2}{970.4} = \frac{1{,}200 - 186}{970.4} = 1.045$$

The feed water temperature has been assumed, 218°F. as before. This is 10° below the temperature of the auxiliary exhaust which is used for feed heating. In calculating the factor of evaporation, the initial condition has been assumed as dry steam, no account being taken of the 60° superheat. It is planned in this case to use a waste heat type superheater, installed in the uptake. All the data which have been used in this estimate are based upon Scotch boilers without superheaters, and obviously a superheater absorbing heat from the waste gases does not influence the boiler performance. If a flue type superheater were used, the heat of the superheated steam would have to be considered.

If we assume the heat contents of the coal as 14,000 b.t.u. per lb. and the boiler efficiency 67 per cent we have:

$$\text{Coal per hour} = \frac{47{,}300 \times (1{,}200 - 186)}{14{,}000 \times .67} = 5{,}120 \text{ lbs.}$$

$$\text{Grate surface} = \frac{5{,}120}{22} = 232 \text{ sq. ft.}$$

Allowing a tentative ratio of $\frac{\text{H.S.}}{\text{G.S.}}$ of 36, we have:

$$\text{Heating surface} = 232 \times 36 = 8{,}350 \text{ sq. ft.}$$

Actual evaporation per sq. ft. H.S. $= \frac{47,300}{8,350} = 5.68$ lbs. per hour.

Equivalent evaporation per sq. ft. H.S. $= 5.68 \times 1.045 = 5.94$ lbs. per hour.

Actual evaporation per lb. of coal =

$$\frac{47,300}{5,120} = \frac{(14,000 \times .67)}{1,014} = 9.25 \text{ lbs.}$$

Equivalent evaporation per lb. of coal $= 9.25 \times 1.045 = 9.70$ lbs.

In Fig. 113 is a curve showing average results from the performance and trials of a large number of Scotch boilers. It will be observed from these curves that at a rate of combustion of 22 lbs. per sq. ft. the equivalent evaporation is 6.2 lbs. per sq. ft. H.S., which is slightly higher than our estimate. The evaporation per sq. ft. H.S. for a given rate of combustion depends on the ratio of $\frac{\text{H.S.}}{\text{G.S.}}$, so this curve only serves as a rough check.

$$\text{Coal per I.H.P. per hour} = \frac{5,120}{2,900} = 1.76 \text{ lbs.}$$

Allowing, as before, an increase of 15 per cent in heating surface to cover all contingencies, such as poor firing, inferior coal, dirty boilers and increased steam consumption, we have:

Heating surface to be provided $= 8,350 \times 1.15 = 9,600$ sq. ft.
Grate surface to be provided $= 232 \times 1.15 = 267$ sq. ft.

Evaporation per sq. ft. H.S. in service $= \frac{47,300}{9,600} = 4.93$ lbs. per hour

Allowing 3 boilers of 3,200 sq. ft. we have the following possibility (Sterling's Handbook, p. 363):

Length, 11 ft. 6 in.
Diameter, 16 ft. 6 in.
Furnaces, 4.
Tubes, 414
H.S., 3,072
G.S., 83.2

$\frac{\text{H.S.}}{\text{G.S.}}$, 36.9

Weight 3 boilers (wet) = 640,000 lbs.
Weight per sq. ft. h.s. = 69.5 lbs.

(Compare the above weight with that of the water-tube boilers of Case I).

Considerable discretion based on experience must be used in fixing the value of H.S./G.S. With a large ratio, the gases have a better opportunity to give up their heat to the water. For forced draft a higher ratio should be used than with natural draft. When waste-heat superheaters are to be used a lower value can be used, for heat not taken up in the boiler will be absorbed by the superheater. For the same reason a boiler fitted with a waste-heat superheater can be shorter and forced at a higher rate than one without.

Good practice generally limits the evaporation in scotch boilers to about 5 lbs. per sq. ft. of heating surface per hour. Rates higher than this are liable to cause leaks and shorten the life of the boiler. The above service rates of 5.7 lbs. per sq. ft. of heating surface per hour as a maximum and 4.9 lbs. as a minimum are certainly as high as it is advisable to go.

The results obtained in the three cases above for boiler heating surface are given merely to illustrate a rational method of obtaining the heating surface. The calculations, of course, are subject to considerable modification, depending on the particular type of boiler used and the exact data for this boiler that may be available from trials and experience in service.

The above clearly shows that arbitrarily basing the boiler heating surface on the horsepower of the main engines is not a correct or logical procedure, unless all facts regarding type of boiler, fuel, steam consumption of engines and auxiliaries are given proper weight.

**191. Approximate Draft Required.**—The total draft should be sufficient to overcome the losses through the grate, boiler tubes, flues, superheater, and air heater. Curves showing the necessary draft between furnace and ashpit are given in Fig. 38

| | |
|---|---|
| Between furnace and ashpit | .30 in. |
| Boiler tubes | .40 in. |
| Superheater | .20 in. |
| Howden heater | .30 in. |
| Total draft between ashpit and uptake | 1.20 in. |

The theoretical amount of air required for combustion is approximately 12 lbs. per lb. of coal (157 cu. ft.). Allowing for 50 per cent excess air we have:

$$\text{Cu. ft. of air per min.} = \frac{5{,}120 \times 157 \times 1.50}{60} = 20{,}000$$

Therefore a fan should be provided that will deliver 20,000 cu. ft. of air per min. at a pressure of 1 1/4 in. of water (3/4 oz.). No allowance has been made for the draft created by the stack. This had best be used to provide a margin against any unforeseen contingencies.

**192. Superheater (Waste heat type).**—The following calculation is given as an approximate check on the degree of superheat:

Temperature of gases leaving boiler = 650°F.
Temperature of gases leaving superheater = 550°F. (100° above steam temp.).
Weight of air per lb. of coal with 50 per cent excess air = 18 lbs.
Approx. weight of flue gases per hour (neglecting products of combustion) $= 18 \times 5{,}120 = 92{,}500$ lbs.
Available b.t.u. in waste gases

$$= W_g \times (t_1 - t_2) \times \text{sp. heat} = 92{,}500 \times 100° \times .238 = 2{,}200{,}000 \text{ b.t.u.}$$

B.t.u. available for superheating, per lb. of steam

$$= \frac{2{,}200{,}000}{47{,}300} = 46.5$$

Taking the specific heat of superheated steam as .57 we have as the degrees of superheat possible

$$\frac{46.5}{.57} = 81.5°.$$

The above calculation indicates that 60° superheat can be safely counted on with this installation.

**193. Lubricating Oil Cooler and Pumps.**—For the geared turbine installation we will adopt a gravity system for lubricating the gears and bearings, similar to that shown in Figs. 66 and 67. The system should be capable of lubricating the gearing and carrying away the heat generated in the gears and bearings.

If we assume the efficiency of the gearing and bearings as 93 per cent, which is somewhat worse than good practice, the lubricating oil will have to carry away,

$$\frac{2{,}700 \times .07 \times 2{,}545}{60} = 8{,}000 \text{ b.t.u. per min.}$$

Assume the following conditions for the coolers:

$t_1$ = Temperature of circulating water inlet = 60°F.
$t_2$ = Temperature of circulating discharge = 90°.
$t_3$ = Temperature of lubricating oil entering cooler = 120°.
$t_4$ = Temperature of lubricating oil leaving cooler = 100°.

Lubricating oil required per minute

$$= \frac{\text{b.t.u. per min.}}{(t_3 - t_4) \times \text{sp. heat of oil}} = \frac{8{,}000}{(120 - 100) \times .48} = 832 \text{ lbs. per min.}$$

Allowing the weight of oil as 7.3 per gallon we have:

$$\frac{832}{7.3} = 114 \text{ g.p.m.}$$

The combined suction and discharge heads for the lubricating oil pumps when the gravity tanks are located 30 ft. above the turbine shaft would be about 40 ft.

To this head should be added the velocity head of the oil and the losses due to pipe friction bends and valves, and the loss through cooler.

| | |
|---|---|
| Velocity head of oil at 3 ft. per sec.................. | negligible |
| Loss head pipe friction bends and valves (with oil velocity of 3 ft. per sec.) approx.................. | 20 ft. |
| Loss through cooler (10 lbs./sq. in.)................. | 27 ft. |
| Suction and discharge head ....................... | 40 ft. |
| Total head required at pump....................... | 87 ft. |

Hp. of lubricating oil pump at 60 per cent efficiency

$$= \frac{832 \times 87}{33{,}000 \times .60} = 3.7 \text{ hp.}$$

$$\text{Cooling water required} = \frac{\text{b.t.u. to remove per min.}}{t_2 - t_1}$$

$$= \frac{8{,}000}{90 - 60} = 267 \text{ lbs. per min.}$$

As already pointed out, the pressure of the oil in the cooler should be greater than that of the water so that if there is any leakage it will be the oil leaking into the circulating water. This requires that the cooler be designed so that the resistance of the circulating water is less than 10 lbs. per sq. in. Allowing 21 ft. as the total head required at the pump, we have:

Horsepower of circulating pump at 50 per cent efficiency

$$= \frac{267 \times 21}{33{,}000 \times .50} = 0.34 \text{ h.p.}$$

As the sea water may at times reach a temperature much higher than 60° the rise in temperature of the cooling water may be only 10° instead of the 30° allowed. Under these conditions 1.00 hp. would be required. A pump and motor capable of this increased duty should be installed.

In this installation it is proposed to use motor driven pumps for both oil and cooling water. A direct acting lubricating oil pump should be installed as a stand-by.

**194. Fuel Oil System.**—In order that the fuel oil will have a low viscosity so that it will atomize in the burners it must be heated to between 100° and 240°F., depending on the kind of oil used.

With following conditions existing we can now calculate the amount of steam from the auxiliary line required for heating the fuel oil.

| | |
|---|---|
| Fuel oil required per hour (see boiler calculations) | = 2,620 lbs. |
| Temperature of oil in tanks (assumed) | 60°F |
| Temperature of oil at burners | 200°F. |
| Specific heat of oil | 0.50 |
| Efficiency of heater, per cent | 85 |
| Temperature of drip leaving heater | 250° |
| Heat of vaporization at 100 lbs. gage | 880 b.t.u. |
| Heat in liquid above 250° | 90 b.t.u. |

Heating steam required

$$= \frac{2{,}620 \times .50 \times (200 - 60)}{(880 + 90) \times .85} = 222 \text{ lbs. per hour.}$$

When the temperature of the oil in the fuel oil tanks is too low to be handled by the pumps a heating coil has to be fitted

at the end of the suction line. The efficiency of such a coil will be much lower than the heater. In view of these possibilities, 300 lbs. of steam is a fair allowance for heating the fuel oil.

**195. Fuel Oil Pumps.**—Two oil pumps are required: one, the booster or transfer pump, draws the oil from the tanks and delivers it to the fuel oil tanks amidships or settling tanks in the fire room; the other, the fuel oil service pump, takes the oil from the discharge of the booster pump and discharges it through the heaters to the burners. The service pumps are always fitted in duplicate, one serving as a stand-by pump.

The exact division of the total head between these pumps varies with different installations. For our present purpose, where the steam necessary is all that is required, the horse power of the two will be combined.

The oil pressure necessary at the burners varies between 100 and 200 lbs. per sq. in.—160 lbs. per sq. in. will be allowed in this case.

| | Feet |
|---|---|
| Head required at burner (160 lbs./sq. in.) | 430 |
| Suction and discharge heads | 10 |
| Pipe friction, head lost in bends, valves, strainers, manifolds, etc. | 70 |
| Head lost through heater | 20 |
| Total head required for pumps | 530 |

Fuel oil per hour = 2,620 lbs.

$$\text{Oil h.p. required} = \frac{2{,}620 \times 530}{60 \times 33{,}000} = 0.70 \text{ hp.}$$

Allowing say 1.0 h.p., required in the oil cylinders of the pumps, we have

$$\text{Steam required per hour} = 100 \times 1.0 = 100 \text{ lbs.}$$

(The steam consumption of direct acting pumps is generally figured in lbs. per water horsepower per hour and varies between 100 and 125 lbs. per w.h.p. per hour.)

**196. Feed and Filter Tank.**—The feed and filter tank should have sufficient capacity to hold at least fifteen minutes' supply of feed water.

In this case we have:

$$\text{Capacity of tank} = \frac{33{,}000}{4 \times 62.5} = 132 \text{ cu. ft.}$$

The shape of the tank can be made to fit its location and should have a total capacity at overflowing of about 150 cu. ft. About one-sixth the volume of the tank is used as a filter tank. With reciprocating engines when there is a considerable amount of oil in the exhaust steam the filter tank becomes an important unit. Generally with reciprocating engine installations grease extractors are fitted on the feed line between the feed pump and heater or between the hot well pump and heater when a hot well pump is used. However, improvements such as the "steam seal" on the L.P. rod and special attention to the extraction of oil from the auxiliary exhaust drip from the feed heater have reduced the amount of oil in the feed. With a turbine installation, and especially when turbine and electric driven auxiliaries are used, the filter tank becomes of much less importance.

**197. Condenser Design:**

Barometer = 30 in.

Turbine designed for a vacuum of 28.8 in. mercury.

Drop in pressure between exhaust nozzle and air pump suction = 0.2 in.

Condenser vacuum = 28.8 + .20 = 29 in.

Steam per hour through condenser = 27,000 lbs.

Temperature corresponding to 29 in. = 79°F.

Temperature of circulating water inlet ($t_1$) = 55°F.

Temperature of circulating water discharge ($t_2$) = 67°F. (79° − 12°).

The condenser should be designed so that the temperature of the circulating discharge is *at least* 10° below the temperature of the steam in the condenser to allow for reduced hot well temperature as discussed under air leakage. (The type of air pump to be used will generally fix the temperature at which the condensate should be withdrawn.)

An allowance of 10 cu. ft. of free air per min. has been made for air leakage. This is rather high for good practice but it is not safe to be too optimistic; it should be possible to maintain the required vacuum of 29 in. with an air leakage somewhat greater than that of good practice. For the same reason a

low value of the air richness ratio $(P_s/P_t)$ has been chosen and a cleanliness factor of .90 assumed.

10 cu. ft. per min. = 600 cu. ft. per hour = 45.5 lbs. free air per hour.

Ratio air to steam by weight at entrance of condenser

$$= \frac{45.5}{27,000} = .0017$$

Thus every pound of steam contains .0017 lb. of air. Under these conditions the partial pressure of the air is negligible at the *top* of the condenser.

(*a*) *Circulating Water.* Salt water is circulated through the condenser to condense the steam by carrying away the heat of vaporization. The formula for heat transfer in b.t.u. per sq. ft. per degree difference per hour, is

$$U = 350\ C_1C_2\ (P_s/P_t)^2\ \sqrt{V} \text{ (Art. 167)}$$

$C_1 = .90$ for cleanliness factor.
$C_2 = .98$ for admiralty tubes.
$(P_s/P_t) = .90$ (assumed)
$(P_s/P_t)^2 = .81$
Hot well temperature (air pump suction) = 67°
$U = 350 \times .90 \times .98 \times .81 \times 2.83 = 705$ b.t.u. per degree difference per hr.

The heat $(H)$ to be removed by the cooling water is the heat of vaporization in the exhaust steam at 29 in. vacuum.

Actually, $H = (H_2 - q_3) \times$ steam per hour $= (x_2r_2 + q_2 - q_3) \times$ steam per hour, where $H_2$ is total heat at vacuum temperature and $q_3$ is heat of liquid at hot well temperature.

For the turbine under investigation the quality after adiabatic expansion from 275 lbs. and 100° superheat to 29 in. was 0.77. In the actual turbine, due to friction, etc., the quality would be about .92. In order to be on the safe side we will assume dry steam at exhaust. $H_2$ at 79° = 1,094.2, $q_3$ at 67° = 35.1, $H_2 - q_3 =$ 1,059.1.

$H = 1,059 \times 27,000 = 28,600,000$ b.t.u. per hr. to be removed by cooling water.

Quantity of cooling water ($Q$) per hr. in lbs.

$$= \frac{\text{B.t.u. to be removed per hour}}{t_1(\text{exit}) - t_2(\text{inlet})}$$

$Q = 28{,}600{,}000/(67-55) = 2{,}380{,}000$ lbs./hr.
$Q = 2{,}380{,}000/(60\times 8.55) = 4{,}630$ g.p.m.
Ratio of cooling water to condensed steam $(Q/W) = 2{,}380{,}000/27{,}000 = 88$.

It is interesting at this point to note the large amount of cooling water required with high vacua. Due to the low temperature existing in the bottom of the condenser with 29 in. vacuum, namely 67°, and the lowest advisable temperature to use for the sea water (55°), there is a difference in temperature of only 12° between the inlet and discharge of the cooling water. With a vacuum of 25 1/2 in. the temperature in the bottom of the condenser would be about 118° (130−12). Using the same sea temperature this would give a temperature difference of 63° between inlet and discharge which would require only one-fifth as much cooling water for the same b.t.u. per hour. We thus see that the cost of producing a vacuum of 29 in. requires a very large increase in size and power of circulating pump. Later on we will see that the sizes of the condenser and air pump are also greatly increased for high vacua.

(*b*) *Cooling Surface.* The mean temperature difference between the steam and water ($t_m$) is,

$$t_m = \frac{t_3 - t_4}{\log_e \frac{t_3}{t_4}} = \frac{24 - 12}{\log_e 2} = 17°$$

where $t_3$ = initial temperature difference (79−55)
$t_4$ = final temperature difference (79−67)

Generally an arithmetic mean can be used without any appreciable error. In this case the arithmetic mean is

$$79 - \frac{(67+55)}{2} = 18°.$$

Due to the fact that the temperature within the condenser is less at the bottom than at the top (in this case 12° lower), the above method of arriving at a mean temperature difference is in error. However, the constant 350 was found from condenser

tests using this method of arriving at the mean temperature; hence the results will be consistent.

$$\text{Cooling surface required} = \frac{28{,}600{,}000}{705 \times 17} = 2{,}400 \text{ sq. ft.}$$

Allowing 10 per cent for margin we have c. s. = 2,640 sq. ft.

$$\text{Condensation per sq. ft. c.s.} = \frac{27{,}000}{2{,}400} = 11.25$$

$$\text{Cooling surface per s.h.p.} = \frac{2{,}640}{2{,}700} = .98 \text{ sq. ft.}$$

(*c*) *Condenser Tubes.* Two sizes of tubes are in common use 3/4 in. & 5/8 in. o.d. Calculations will be carried through for both sizes. The 3/4 in. tubes give a much more satisfactory condenser than the 5/8 in. tubes on account of the smaller number of tubes required. The 5/8 in. tubes, however, give a lighter condenser. Tubes of 3/4 in. diameter are used as a rule for condenser on merchant ships and 5/8 in. diameter on naval vessels. In land installations where the circulating water is liable to be dirty 1 in. and sometimes 1 1/4 in. tubes are used.

Using 5/8 in. tubes (18 B.W.G.):
o.d. = .625 in.
i.d. = .537 in.
Outside surface per lineal ft. = .164 sq. ft.
Area through one tube = .00157 sq. ft.
Quantity of water per sec. = 10.45 cu. ft. (4,700 g.p.m.)
Velocity of water through tubes = 8 ft. per sec.
Cooling surface = 2,640 sq. ft.

$$\text{Number of tubes required} = \frac{10.45}{.00157 \times 8} = 832$$

$$\text{Length of tube} = \frac{\text{cooling surface in sq. ft.}}{\text{No. of tubes} \times \text{surface area per ft.}} = \frac{2{,}640}{832 \times .164}$$

Length = 19.35 ft. for one pass condenser (832 tubes)
= 9.67 ft. for two pass condenser (1,664 tubes)

Using 3/4 in. tubes (18 B.W.G.):

o.d. = .75 in.
i.d. = .652 in.

Outside surface per lineal ft. = .196 sq. ft.
Area through one tube = .00232 sq. ft.

$$\text{Number of tubes required} = \frac{10.45}{.00232 \times 8} = 564$$

$$\text{Length of tube} = \frac{2,640}{564 \times .196} = 24 \text{ ft.}$$

Length = 24 ft. for one pass condenser (564 tubes)
= 12 ft. for two pass condenser (1,128 tubes)

A layout can now be made for the condenser with the proper length, area of tube sheet, etc. Experience with the details of condenser construction is, of course, necessary in order to draw up a suitable condenser.

(*d*) *Vacuum in the Tropics.* To illustrate the loss in vacuum that a ship will experience in tropical waters, the following calculation is given:

$t_1$ (sea temperature) = 75°

Condenser temperature possible with circulating water leaving condenser at a temperature 12° below that of steam in condenser and with a 10° rise in cooling water:

75 + 10 + 12 = 97° (vacuum corresponding = 28 1/4 in.)
Temperature of inlet water = 75°
Temperature of exit water = 85°
Temperature of top of condenser = 97°

With this falling off in vacuum there will be a reducing in s.h.p. of the engines and an increase in the steam consumption per s.h.p. If, however, we allow the same total steam as before, we have:

$$t_m = \frac{(97 - 75) - (97 - 85)}{\log_e 1.83} = \frac{10}{.604} = 16.5°$$

Both $U$ and $t_m$ are practically the same as before; so the cooling surface is sufficient.

The heat to be removed = $(H_2 - q_{85}) \times 27{,}000$
= 1,049.3 × 27,000 = 28,300,000 b.t.u. per hour.

$$Q = \frac{28{,}300{,}000}{(85 - 75) \times 60 \times 8.55} = 5{,}500 \text{ g.p.m.}$$

This brings out the fact that the circulating pump should have a reserved capacity over that calculated to allow for tropical conditions and also for possible increases in steam consumption over that allowed.

A vacuum somewhat higher than 28 1/4 in. could be produced with sea water at 75° if a larger quantity of cooling water were used. This, however, would increase the velocity of the water through the tubes and call for a pump delivering not only a larger quantity of water but at a greater pressure due to the increased head caused by the higher speed of the water through the tubes.

**198. Condensers for Low Vacua.**—For low vacuum condensers where the rise in the temperature of the cooling water between inlet and discharge is large, the speed of the cooling water will have to be reduced in order to keep the length of the condenser down. With a high velocity of the cooling water, the number of tubes required to give the proper area to pass the cooling water through ($Q = av$) is greatly reduced; and in order to get the required cooling surface with the smaller number of tubes long tubes would have to be used.

Speeds of 3 to 5 ft. per second should be used with installations of this type and the discharge temperature of the cooling water should not be allowed to exceed much over 100°F. As already pointed out, temperatures higher than this will cause trouble by scale forming on the tubes. An example of an extreme case will make this clear:

Vacuum = 26 in. (125°F.)
Inlet temperature = 55°F.
Discharge temperature = (125 − 12) = 113°.
Steam condensed per hour = 46,400 lbs.
B.t.u. to be removed per hour = 48,000,000 = (1,033 × 46,400).
Cooling water per hour =

$$\frac{48{,}000{,}000}{113 - 55} = 827{,}500 \text{ lbs. per hour (3.59 cu. ft./sec.)}$$

Cooling surface required = 2,268 sq.ft. (allowing 10 per cent excess as before).
3/4 in. tubes (18 B.W.G.)
Cross sectional area (inside) = .00232 sq.ft.
Cooling surface per foot of length = .196 sq.ft.

Number of tubes in one pass =

$$\frac{3.59 \text{ cu. ft. per sec.}}{.00232 \times 8} = 194 \text{ tubes with velocity of 8 ft./sec.}$$

$$\text{Length of two pass condenser} = \frac{2{,}268}{194 \times .196} \times 1/2 = 30.00 \text{ ft.}$$

If speed of cooling water is taken as 4 ft. per second, we have:

$$\text{Length} = \frac{3{,}200}{388 \times .196} \times 1/2 = 21 \text{ ft. for two pass}$$

A reduction in the speed of the cooling water decreases $U$ and hence increases the cooling surface as the square root of the change in velocity, and increases the number of tubes directly as the change. Hence the net result is that the length of the condenser changes as the square root of the change in velocity

**199. Condenser for Reciprocating Engine Installation.—**

Vacuum in condenser = 27 in. (115°F.)
Steam condensed per hour = 40,600 lbs.
$t_1$ (at entrance) = 55°F.
$t_2$ (at exit) = 90°F. (115 − 25°).
V = 4 ft. per sec.

Temperatures of sea water greater than 90 or 100°F. are liable to cause scale unless the speed of the circulating water is sufficiently high.

$U = 350 \times .90 \times .98 \times .81 \times 4 = 500$ b.t.u. per degree difference per hour.

Hot well temperature (air pump suction) = 105°F. (115 − 10°).

$$H_2 - q_{105} = 1{,}110 - 73 = 1{,}037.$$

$$H = 1{,}037 \times 40{,}600 = 42{,}150{,}000 \text{ b.t.u. to remove per hour.}$$

$$Q = \frac{42{,}150{,}000}{(90 - 55) \times 60 \times 8.55} = 2{,}350 \text{ g.p.m.}$$

$$t_m = \frac{(115 - 55) - (115 - 90)}{\log_e 60/25} = \frac{35}{.875} = 40°$$

$$\text{Cooling surface} = \frac{42{,}150{,}000}{500 \times 40} = 2{,}110 \text{ sq. ft.}$$

Allowing 10 per cent margin, c.s. = 2,320 sq.ft.
Cubic feet of water through tubes per second = 5.23.

Number of 3/4 in. (18 B.W.G.) tubes per pass $= \frac{5.23}{4 \times .00232} = 562.$

Length of tubes for two pass condenser =

$$\frac{2{,}320}{562 \times .196} \times 1/2 = 10.5 \text{ ft.}$$

The above calculation brings out clearly the smaller quantity of cooling water and reduced condenser cooling surface required for a 27-in. vacuum. Although the steam per hour through the condenser in this case is 50 per cent greater than for the 29-in. vacuum, the amount of cooling water is 100 per cent less. This reduction in the amount of cooling water is of course due to the larger rise in the temperature of the cooling water in its passage through the condenser which is possible because of the higher temperature existing in the condenser.

In the foregoing calculations a temperature of 55°F. has been allowed, for special reasons, for the sea water at entrance to the condenser. This is rather low for the average sea water and entirely too low for summer conditions. For conservative calculations a temperature higher than this should be used.

**200. Circulating Pump (Centrifugal).**—In view of the requirements to maintain a good vacuum in the tropics, the circulating pump should have a capacity of 6,000 g.p.m. and be capable of maintaining a water velocity through the tubes of 9 ft. per second (8 ft. per second is the designed requirement).

The head lost through 30 ft. of 5/8-in. tube with a velocity of 8 ft. per sec. is 22 1/2 ft.[1] This includes losses at entrance and exit of the tubes. To this should be added the lost head in valves, strainer, bends, changes in section, and friction in the piping to and from the sea. The velocity in the piping should be in the neighborhood of 12 ft. per sec. The total required head would thus probably be around 30 ft. in service. Allowing a 20 per cent increase in head for contingencies, fouling, etc., we have with 65 per cent efficiency of pump:

$$\text{Required hp.} = \frac{6{,}000 \times 8.55 \times 30 \times 1.20}{33{,}000 \times .65} = 86 \text{ hp.}$$

[1] See Sterling's "Marine Engineers' Handbook," Section 8.

$$\text{Service hp.} = \frac{4{,}700 \times 8.55 \times 30}{33{,}000 \times .65} = 56 \text{ hp.}$$

**201. Condensate Pump.**—If an injector type of air pump is used, as the "Radojet" which is used in this case, a condensate pump must be installed to handle the condensed water from the condenser and the condensed steam used by the first stage of air pump. (See description of "Radojet" air pump with intercondenser.)

This pump will have to raise the water from a pressure corresponding to 29 in. to atmospheric pressure, or 33 ft. Allowing for pipe friction, entrance losses, etc., and a small lift to feed tank, the total head will probably be around 40 ft.

Water handled per hour, 27,000+900=27,900 lbs.

Allowing 60 per cent as the efficiency of the pump we have:

$$\text{Hp.} = \frac{27{,}900 \times 40}{60 \times 33{,}000 \times .60} = 0.94. \text{ hp.}$$

In this installation we will use a motor driven centrifugal pump.

Hp. input at 77 per cent motor efficiency = 1.2 hp.

**202. Boiler Feed Pump.**—Feed per hour under the best conditions = 33,000 lbs.

The capacity of the pump should be twice the rated consumption so that it can take care of any sudden overload or bring the level of the water in the boiler back to normal quickly in case it should fall for any reason. Boiler feed pumps must be fitted in duplicate, one for a service pump and one for a stand-by. The feed pump must overcome the boiler pressure and the lost heads due to pipe friction, bends, valves, and feed heater and in addition produce a head sufficient to create a velocity of flow into the boiler. In this case where the boiler pressure is 270 lbs., a head of 300 lbs. should be allowed at entrance to the boiler. An estimate of the lost heads for this installation indicates a total loss of about 50 ft. with a possible loss of 100 ft. due to partially closed valves.

In the selection of a boiler feed pump we have a choice between several types of direct acting reciprocating pumps and a centrifugal pump. At high pressures the centrifugal pump gives a much more even delivery and avoids the shock caused by

reciprocating pumps Hence it is easier on the piping, check valves, etc. Also the steam consumption of a direct acting pump is more than twice that of a turbine driven pump; and in this installation it can be foreseen that the auxiliary exhaust steam for feed heating is running high.

A turbine driven centrifugal pump will be selected for the service pump with a direct acting vertical simplex pump as a stand-by. An efficiency of 65 per cent can be counted on for a centrifugal type of pump and efficiencies better than this may be expected.

300 lbs. sq.in. = a head of $300 \times 2.30 = 690$ ft.

$$\text{Max. hp.} = \frac{33{,}000 \times 2 \times 790}{33{,}000 \times 60 \times .65} = 40.5 \text{ hp.}$$

$$\text{Service hp.} = \frac{33{,}000 \times (690 + 50)}{33{,}000 \times 60 \times .65} = 19 \text{ hp.}$$

Required pump: 130–65 g.p.m.; head 790 ft.

This will probably require a three stage pump.

**203. Evaporator.**—An allowance should be made for a make-up feed in service of 5 per cent which in this case is 1,400 lbs./hour. This value is much less than common practice but is none too low for a good installation using superheat steam where careful attention is given to small leaks. The capacity of the evaporator should be somewhat greater than twice this allowance or about 30 tons per 24 hours. Most cargo ships that are fitted with evaporators have single effect evaporators. For the best economy, however, only double effect evaporators should be considered. The following rates for steam used by evaporators are taken from the Journal of the A. S. N. E., Feb., 1914:

1 lb. steam at 50 lbs. will evaporate (single effect) 0.8 lb. water per hour.
1 lb. steam at 50 lbs. will evaporate (double effect) 1.45 lbs. water per hour.
1 lb. steam at 50 lbs. will evaporate (triple effect) 1.95 lbs. water per hour.

If a triple effect evaporator is fitted with evaporator feed heater using steam from third effect the evaporation will be 2.35 lbs.

water per hour per pound of steam. With a double effect evaporator 1 lb. steam at 50 lbs. gage will evaporate about 1.45 lbs. of water. Allowing this evaporation, 1,400 lbs. make-up feed will require:

$$\frac{1,400}{1.45} \text{ lbs.} = 970 \text{ lbs. of steam at 50 lbs. gage.}$$

**204. Auxiliaries.**—In the merchant service the auxiliaries are supplied by a dry steam line which leaves the boilers before the steam has been passed through the superheater. In some ships, especially in the U. S. Navy, full boiler pressure is used on all the auxiliaries. The better practice is to use reduced pressure on the auxiliaries except direct acting boiler feed pumps. In this case where a centrifugal feed pump is used reduced pressure can be used on all the auxiliaries. Reducing the pressure, which is a constant heat contents change, results in superheating the steam and thus avoids the losses due to condensation and drip from pipes, valves, valve chests, etc.

If for the case under consideration the pressure in the auxiliary line is reduced from dry steam at 285 lbs. abs. to 100 lbs. gage, we have from the Mollier diagram:

$$H_1 \text{ at 285 lbs. dry} = 1,203 \text{ b.t.u.}$$

At 115 lbs. abs. and 1,203 b.t.u. (constant heat line) the steam is at about 25° superheat.

All the auxiliaries exhaust to the auxiliary exhaust line at 5 lbs. gage, and the auxiliary exhaust line delivers the steam to the feed heater.

In some installations there is a cross-connection from the superheated steam line to the auxiliary saturated steam line and the auxiliaries are started on saturated steam and then operated with superheated steam. This procedure, of course, could not be used when a reduced steam pressure is used on the auxiliaries. In fact, using superheat with auxiliaries has very little to recommend it.

We are now ready to tabulate our auxiliaries and make an estimate of the steam used by each. The following steam consumptions have been allowed and are representative figures

### ESTIMATE OF STEAM CONSUMPTION FOR AUXILIARY MACHINERY 2,700 S.H.P. GEARED TURBINE WITH OIL FIRED BOILERS

| Auxiliary | Type of engine | Type of aux. | Service hp. | Steam per hp. hr. | Total steam per hr. | Remarks |
|---|---|---|---|---|---|---|
| Circulating pump | turbine | centrif. | 56.0 | 30 | 1680 | |
| Air pump | Radojet | ........ | ........... | ...... | 900 | Exhausts into feed tank |
| Condensate pump | motor | centrif. | 1.0 | [1] | [1] | |
| Boiler feed pump | turbine | centrif. | 20.5 | 40 | 820 | |
| Blower | recip. | fan | 4.0 | 60 | 240 | |
| Lub. oil pump | motor | centrif. | 3.7 | [1] | [1] | |
| Fuel oil pumps (2) | direct-acting | | 1.0 | 100 | 100 | Transfer and service pumps |
| Lub. oil cooler pump | motor | centrif. | 0.34 | [1] | [1] | |
| Refrigerating set | motor | ........ | 2.0 | [1] | [1] | |
| Steering gear | motor | ........ | 3.0 | [1] | [1] | Average continuous hp. = 3.0 |
| Evap. and distiller pumps | motor | centrif. | 1.0 | [1] | [1] | Combined hp. = 1.0 |
| Sanitary pump | direct-acting | | 2.0 | 100 | 200 | |
| Fresh water pump | direct-acting | | 2.0 | 100 | 200 | |
| Generating set | turbine | ........ | 28.0 | 35 | 980 | 30 kw set |
| Aux. gen. set | turbine | ........ | 10 kw. | ...... | ...... | Not operating |
| Lub. oil pump | direct-acting | | $7\frac{1}{2} \times 7 \times 10$ | ...... | ...... | Stand-by |
| Fuel oil pump | direct-acting | | $5\frac{1}{4} \times 3\frac{1}{2} \times 5$ | ...... | ...... | Stand-by |
| Boiler feed pump | direct-acting | | ........... | ...... | ...... | Stand-by |
| Fire pump | direct-acting | | $10 \times 12 \times 10$ | ...... | ...... | Not operating |
| Bilge pump | direct-acting | | $6 \times 5\frac{1}{2} \times 6$ | ...... | ...... | Not operating |
| Aux. air and circ | direct-acting | | ........... | ...... | ...... | Not operating. Combined pump |
| Evaporator drip | | | | | 970 | |
| Fuel oil heater drip | | | | | 300 | |
| Total steam excluding drips and Radojet | | | | | 4,220 lbs. per hour | |
| Total steam including drips and Radojet | | | | | 6,390 lbs. per hour | |

[1] Motor driven: steam accounted for under generating set. The following allowances have been made (see separate calculations for each auxiliary) as the *average continuous* service hp. input to the motor driven auxiliaries allowing 83 per cent motor efficiency:

| | |
|---|---|
| Lubricating oil pump | 4.50 hp. |
| Oil cooler circulating pump | 0.40 |
| Steering gear | 3.60 |
| Condensate pump | 1.20 |
| Refrigerating set | 2.40 |
| Evaporator and distiller pumps (combined hp.) | 1.25 |
| Allowance for lighting | 10.00 |
| Total hp. input to motors | 23.35 |

Allowing 85 per cent as the efficiency of the generator, we have $23.35 \div .85 = 27.6$ hp. as the required hp. output of generating set.

for turbines and reciprocating engines working between 100 lbs. gage and atmospheric exhaust:

Direct acting steam pumps 100–125 lbs. per water hp. per hr.
Geared turbines, 25 h.p., 40–50 lbs. per s.h.p. per hr.
Geared turbines, 100 h.p., 35 lbs. per s.h.p. per hr.
Reciprocating engines, 10 h.p. 60 lbs. per s.h.p. per hr.
Weir air pumps 1 per cent total steam.
"Radojet" air pump (See *A.S.N.E. Journal*, Aug., 1920).
"Radojet" air pump with intercondenser, handling 10 cu. ft. free air per min. at 29 in. = 860 lbs.
"Radojet" air pump without intercondenser, handling 10 cu.ft. free air per min. at 29 in. = 1,700 lbs.

The preliminary estimate for the steam for auxiliaries was 5,400 lbs. per hour (20 per cent). The new figure by independent estimate is 6,390 lbs. per hour or 23.6 per cent of that used by the main engines. The total increase in the steam required is 990 lbs. or practically 3 per cent increase—hardly enough to warrant an increase over our preliminary estimate for heating surface.

**205. Calculations for Feed Heating.—**

*Estimated Steam Consumption:*

| | Lbs. |
|---|---|
| Steam used by main engine per hour | 27,000 |
| Auxiliary exhaust at 5 lbs. gage to feed heater | 4,220 |
| Radojet, evaporator and oil heater drip | 2,170 |
| Steam for heating[1] | 600 |
| Total steam per hour | 33,990 |

*Heat available in feed tank:*

B.t.u. available above 32°F.:

| | | | B.t.u. |
|---|---|---|---|
| In make-up feed (5 per cent) at 200°F. from evaporator[2] | 1,400 × | 168 = | 235,000 |
| In condensate from condenser at 67°[2] | 25,600 × | 35 = | 897,000 |
| In auxiliary exhaust drip from feed heater at 228° | 4,220 × | 196 = | 828,000 |

[1] For convenience, drips from heating steam taken at 125°.

[2] For convenience all the make-up feed assumed to be lost from main engine steam.

| | | |
|---|---|---|
| In evaporator drip at 250°........ | $970\times 218=$ | 212,000 |
| In oil heater drip at 250°......... | $300\times 218=$ | 65,500 |
| In Radojet exhaust[3]............. | $900\times 1{,}000=$ | 900,000 |
| Total b.t.u. in feed tank above 32°F........... | | =3,137,500 |

$$\text{Temperature of water in feed tank} = 32° + \frac{3{,}137{,}500}{33{,}990} = 125°\text{F}.$$

Heat available in a pound of auxiliary exhaust = heat of vaporization at 5 lbs. gage.

Total heat available in auxiliary exhaust $= 959 \times 4{,}220 =$ 4,050,000 b.t.u. per hour.

Allowing an efficiency of 90 per cent for the heater due to radiation, we have for the temperature of the feed water at entrance and exit of feed heater:

Temperature of feed entering heater = 125°F.

$$\text{Rise in temperature in feed heater} = \frac{4{,}050{,}000 \times .90}{33{,}990} = 107°\text{F}.$$

Temperature of feed leaving heater $= 125 + 107 = 232°$.

It will be observed that the calculated temperature of the feed water is 4° higher than the temperature of the heating steam. With a closed heater the feed water can as a rule be heated to within about 10° of the steam temperature. In this case the maximum temperature possible for the feed water would be about 218° (228–10). The foregoing calculations, which show that the temperature is greater than that of the heating steam, of course, simply indicates that there is more heating steam available than can be used.

In a case such as this we have four alternatives: (1) Better designed auxiliaries, using less steam, could be installed; (2) the steam pressure in the exhaust line could be increased, thus increasing the temperature of the steam entering the heater; (3) part of the exhaust steam could be discharged into the low pressure stages of the main turbine; or (4) a series heat could be employed with part of the exhaust steam at a higher pressure.

[3] Radojet fitted with intercondenser but heat accounted for in this manner.

If a separate coil were fitted in the discharge end of the heater supplied with exhaust from the direct acting steam pumps, which discharge at a much higher pressure than the other auxiliaries, a certain amount of steam would be available at a temperature higher than 228°. This would allow the feed to be heated to a higher temperature. The series coil might be connected to the drips from oil heaters, evaporators, etc., which are at a higher temperature than 228°; or the back pressure on certain auxiliaries might be carried higher than 5 lbs. gage and this exhaust sent through the second coil of the feed heater. While the use of a series feed heater offers an opportunity for better economy and a higher feed temperature, it would require two separate auxiliary exhaust lines and otherwise complicate the piping. Hence, this installation would not appeal to many shipowners, and the best solution would be to use the surplus exhaust in the L.P. turbine. In no case should continuous surplus of exhaust steam be sent to the auxiliary condenser.

Ordinarily the drains from the heating system when in operation are led to the feed tank. As the heating steam is at a low pressure and very small in quantity compared to the total feed, no allowance has been made for it in the foregoing calculations.

If the make-up feed had been drawn from the make-up feed water tank in the double bottom instead of being evaporated from salt water our feed temperature would be slightly lower. The general practice in merchant ships is to draw the make-up feed up from the double bottom through a connection to the main condenser. Thus the vacuum in the condenser serves to lift the water from the tank. The make-up feed mixes with the condensate in the bottom of the condenser. While this is a very simple and convenient arrangement, it is not to be recommended in high vacuum installations.

In the foregoing calculations for feed heating, the exhaust from the auxiliaries has been assumed dry and the total heat of vaporization has been assumed available for feed heating. The steam in the auxiliary line is obtained by reducing steam at full boiler pressure (270 lbs. gage) down to 100 lbs. gage which gives the steam in the auxiliary line a superheat between 20 and 30°F. (Art. 10). This amount of superheat is sufficient to allow for all radiation losses and to insure dry steam at the throttle of all auxiliaries. An adiabatic expansion from dry

steam at 115 lbs. absolute to 20 lbs. absolute would give a quality of 90 per cent at exhaust.

Due to losses in the engines and the increased heat contents in the rejected steam the quality at exhaust is always greater than that after an adiabatic expansion. Further, when direct acting auxiliaries are used the steam may be dry or even superheated at exhaust (Art. 21). For the above reasons, the quality in the exhaust line on shipboard can be assumed as 100 per cent without any serious error.

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